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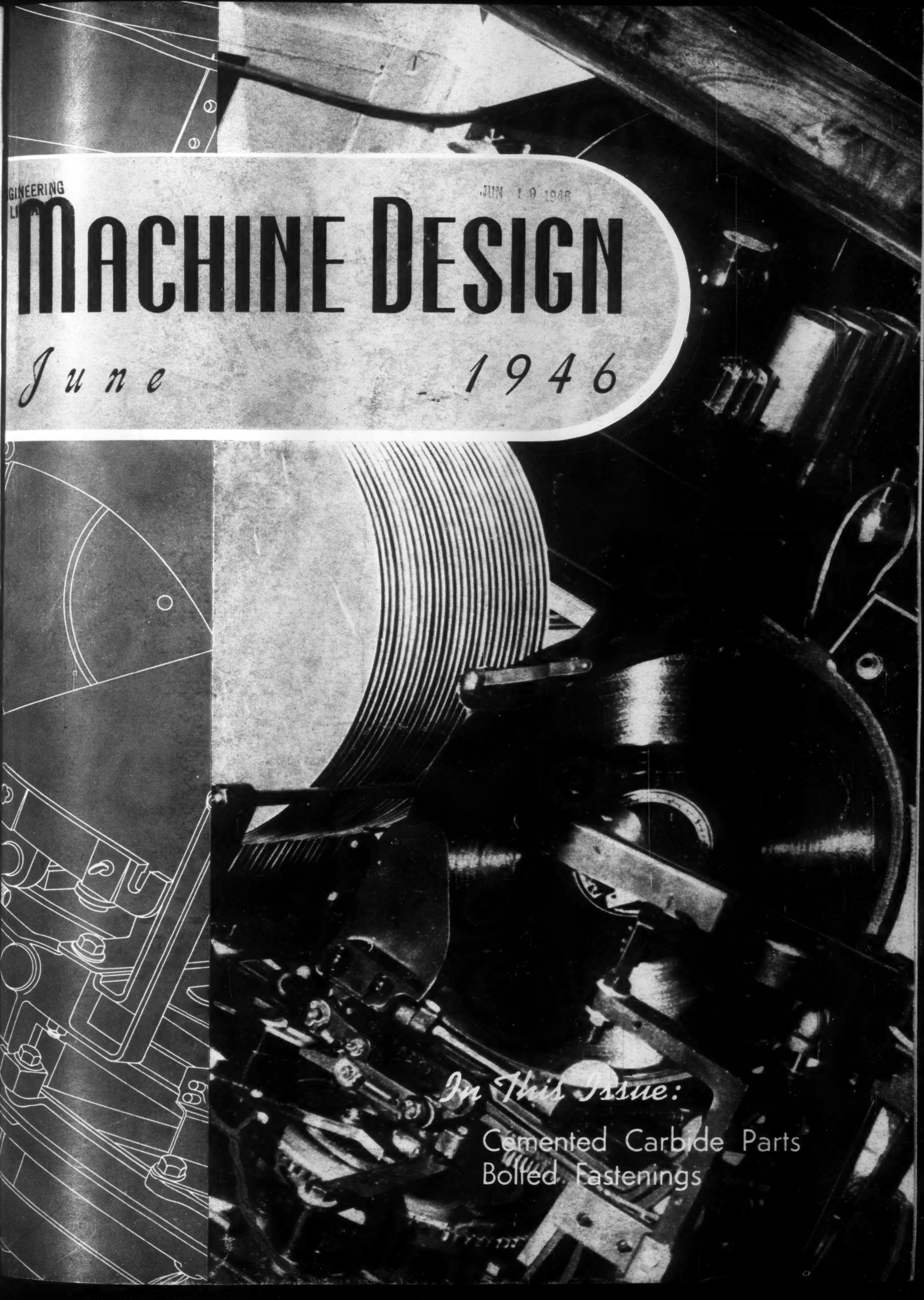
MACHINE DESIGN

June *1946*



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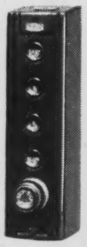

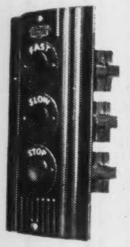

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MACHINE DESIGN
THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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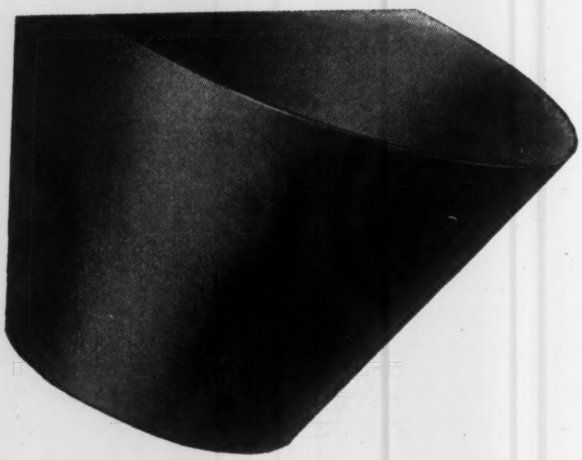
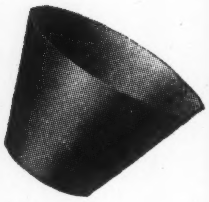


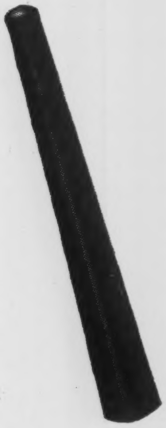


Shaped

TO FIT THE JOB....

Built

TO DO THE JOB....



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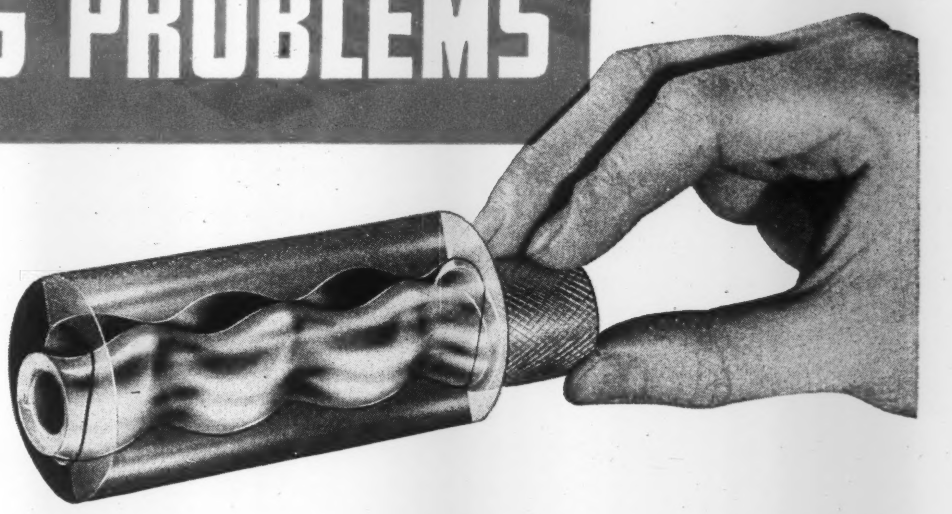
CONTAINERS FOR GASES, LIQUIDS AND SOLIDS

LICKS PUMPING PROBLEMS



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- Brewer's Yeast
- Carbide Sludge
- Cellulose Solutions
- Chemicals
- Citrus Juices
- Clay Slip
- Cold Cream
- Corn Syrup
- Distiller's Slop
- Dog Food
- Enamel
- Fuel Oil
- Gasoline
- Glucose
- Glue, Water Glass
- Greases
- Hash
- Liquefied Gases
- Tool Coolant
- Magnesium
- Marshmallow
- Milk, Cream
- Molasses
- Mud (oil well)
- Oils
- Paper Stock and Coatings
- Paints
- Plastics
- Porcelain
- Pulp (fruit and vegetable)
- Putty
- Resins, hot
- Rubber Cement
- Sand, Silt
- Sludge, Slurries
- Solvents
- Starches
- Tar
- Tooth Paste
- Water (sweet, salt)
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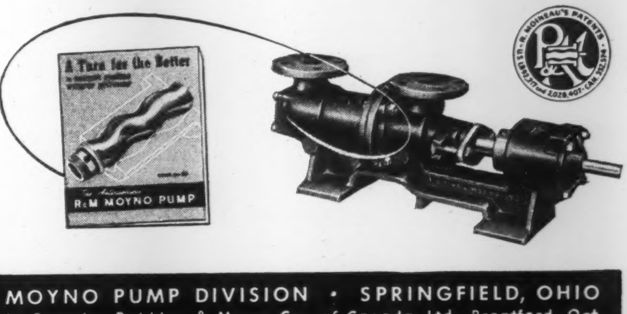

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Topics

SYNTHETIC ALLOYS having physical properties far beyond our day dreams are foreseen as a product of powder metallurgy by Dr. William D. Jones of London, England. In powder metallurgy there are no phase rule limits to what can be put in an alloy as in conventional casting methods. One phase from one alloy, another from a second alloy and so on together with a pinch of some nonmetallic may produce a powder-metal material that had never been possible in a conventional alloy.

PLASTIC PARTS can now be molded in half the time required in prewar years as a result of manufacturing and material improvements. Among these improvements is electronic preheating which alone has resulted in 50 per cent time saving in molding. Plastics of the silicone family promise to surpass other similar materials.

PHENOMENAL GROWTH of the plastics industry during the last five years has placed the United States in the position of the world's largest producer of such materials. From 1939 to 1944 the total dollar volume in output of plastics and synthetic resins increased 325 per cent, accounting for 332 million dollars in 1944.

DEICING STRIP along the leading edges of aircraft wings is one of the most original wartime developments of powder metallurgy applications. It is a porous strip, usually of cupro-nickel, through the pores of which is pumped a deicing fluid in small amounts. Exact quantities of fluid may be pumped to any part of the wing irrespective of section or curvature.

SPEED VARIATION in a range of eight to one for direct-current motors, instead of the conventional four to one range, is possible in a new design utilizing four poles. Two poles are energized conventionally and the other two have controls which vary their flux from maximum to zero and even reverse. Thus the combined flux varies from the sum of the fields through any value down to their difference. Horsepower remains constant for the motor because the armature voltage does not change.

AT LEAST fifty-two electrical devices—not counting lights—can be used in the average home.

ALUMINUM ROLLS for printing cloth in textile mills are being tested at the Swansea Mills, Swansea, Mass. These rolls are lighter than the copper ones generally used. Also, aluminum is being experimented with for rolls in paper printing.

LEAK DETECTOR for vacuum systems utilizes helium gas to ferret out location of any faulty joint in the equipment. Developed by Westinghouse for high-vacuum apparatus used in the atomic bomb project, the detector involves spraying helium gas on any suspected location. This gas is used because it is stable and does not react harmfully with other gases or parts of the system and can be detected readily by a mass spectrometer attached to the system.

AMERICAN MINES will produce less than one-quarter of the silver needed this year by American industries. Total need is estimated at 125 million ounces. According to the Silver Users Emergency Committee, proper legislation and action on pricing should free some idle treasury stock and attract metal from foreign countries.

UTILIZING INFRARED RAYS the Snooperscope and the Sniperscope provide for sight through a telescope in the dark and accounted for many dead Japanese during the war. Suggested peacetime applications include use as crime detection aids and for river boat traffic.

MACHINE DESIGN

Lift Truck Exemplifies Compactness in Design

By Verne Johnson
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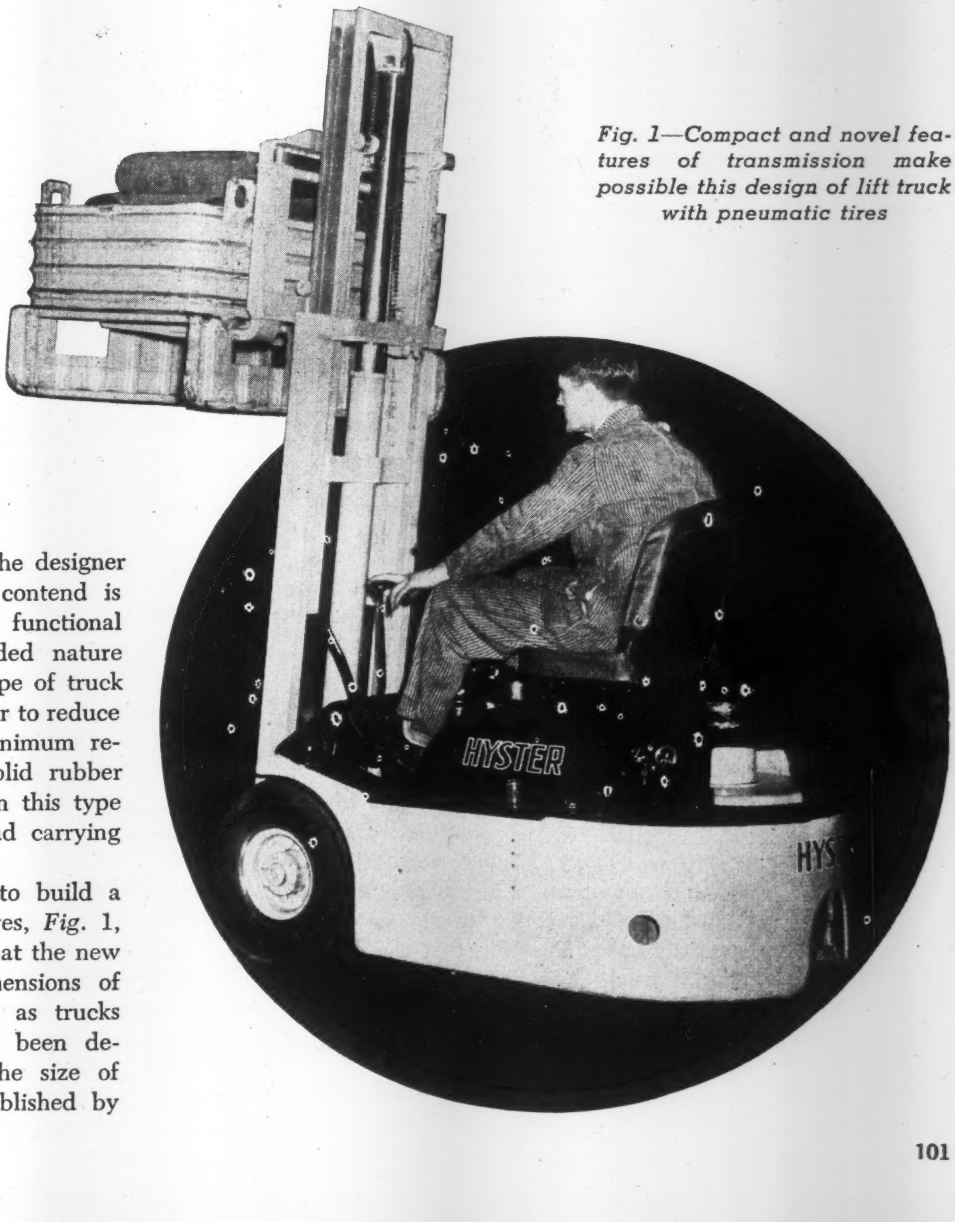


Fig. 1—Compact and novel features of transmission make possible this design of lift truck with pneumatic tires

BIGGEST problem with which the designer of a fork-type lift truck has to contend is lack of space in which to locate functional and operating parts. The crowded nature of the work areas in which this type of truck operates has forced the manufacturer to reduce the outside dimensions to the minimum required by operating stability. Solid rubber tires have been used exclusively in this type of truck because of their high load carrying capacity and small size.

When the decision was made to build a small lift truck with pneumatic tires, *Fig.* 1, it was made with the knowledge that the new truck must have the same dimensions of width, length and turning radius as trucks of comparable capacity that had been designed for solid rubber tires. The size of these trucks had been firmly established by

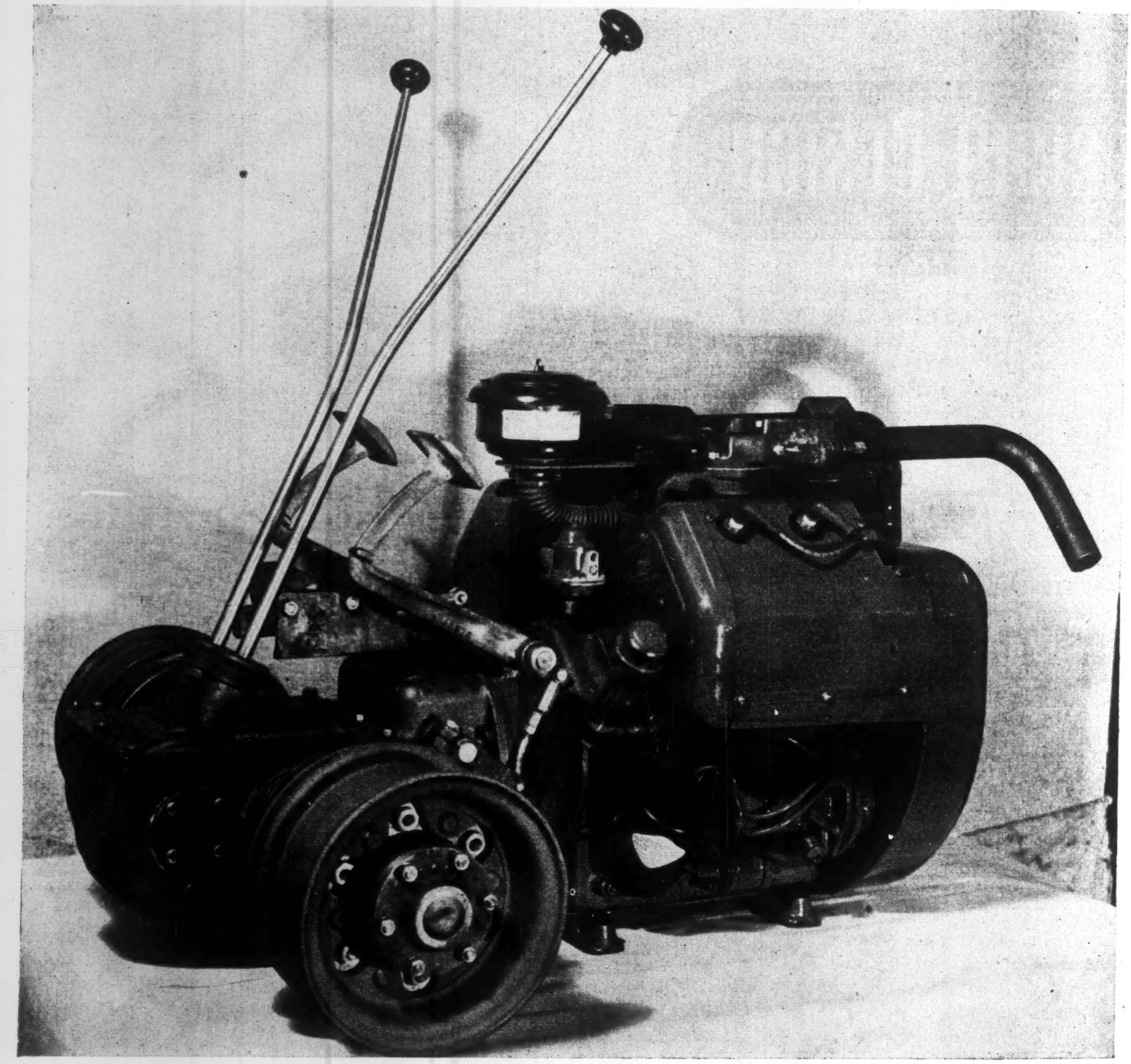


Fig. 2—Power unit is V-type air-cooled engine. Space restrictions require that transmission, differential, clutch and flywheel housing be contained in the $12\frac{3}{8}$-inch distance between engine and front axle

the industry and trade. Any substantial increase in these dimensions would seriously offset the obvious advantages of pneumatic tires and would interfere with sales of the truck even where operating space was not a factor.

As soon as the outside dimensions of the truck were selected a preliminary layout of the design was started. This layout disclosed immediately that the conventional four-wheel lift truck design could not be used. The four solid rubber tires, two driving and two steering, of this conventional design would occupy 13 per cent of the "floor area" whereas the minimum of three pneumatic tires of adequate capacity would account for 35 per cent of the "floor area" of the truck. Knowing that the component parts of previous trucks had been crowded into the smallest practical "package" it was obvious that a radical new design would have to be established to accomplish the desired result.

Another fact that the first layout of the truck disclosed was that it would be impractical to locate any of the heavy functional parts above either the rear steering tire or the front driving tires. In order to safely utilize the higher operating speeds that would be made possible by the smooth riding of pneumatic tires the center of gravity of the truck must be extremely low to prevent overturning on quick turns. Also, the truck would be so small that the operator would ride it as though he were on horseback and the truck would have to be low enough to permit mounting to the seat directly from the floor. These considerations made it necessary to place all of the operating parts of the truck, with the exception of the lifting frames, within the triangle formed by one rear and two front wheels.

Air cooled, V-type, four cylinder engine shown in *Fig.* 2 was selected to power the truck because it was comparatively short and would not require a space consuming radiator for cooling. When this engine was placed in the layout as far to the rear as possible and directly ahead of the rear steering tire it was noted that there were just twelve and three-eighths inches of lengthwise space between the forward face of the engine crankcase and the center line of the front driving axle. This space must contain the engine flywheel, the flywheel housing, the clutch and clutch throwout parts, the full reversing two-speed transmission and the differential. It must also contain the hydraulic tilting mechanism of the truck and the operating control levers for the clutch, brakes, throttle and transmission. It was apparent that it would be impossible to employ separate transmission and differential assemblies with flexible connecting parts, so a design was started to provide the propelling parts and controls in a self-contained single unit. The resulting power unit assembly is illustrated in *Fig.* 2.

Utilizes Unique Transmission

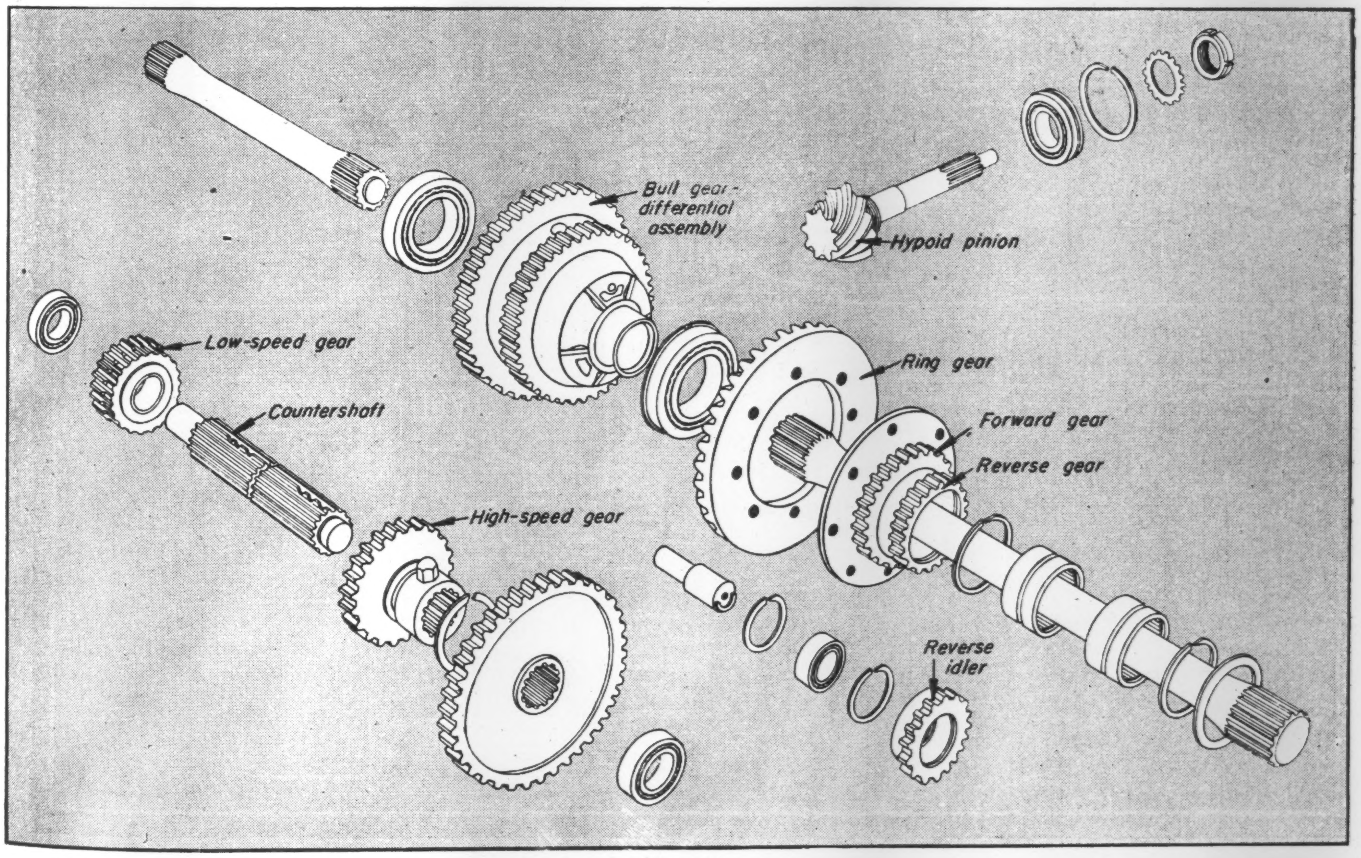
Steps leading to the final design of the transmission, the conception and progressive development of ideas that were tried, purchasing, foundry and machine shop problems, the effect of other functional parts of the truck on the transmission design and the multitude of other problems that required solution are beyond the scope of this discussion. The following will be confined primarily to the major details of the transmission design that was finally adopted for use in this truck.

Power is taken from the flywheel through a conventional, automotive, single plate dry-disk clutch, utilizing a conventional throwout bearing and sleeve assembly to disengage the clutch. The main drive gear, integral with the clutch driveshaft, is a hypoid pinion *Fig.* 3. Hypoid design of drive pinion and ring gear raises the engine above the center line of the differential and drive wheels to provide sufficient running clearance under the engine crankcase. The first drive gear or ring gear is bolted to a flange provided on one end of a two-gear cluster. One of the gears on this cluster is for forward travel and the other provides a reverse through a constant mesh reverse idler gear.

The ring gear and cluster gear are located on the center line of the driving wheels and the hub of the cluster gear has a large bore to permit the left-hand axle driveshaft to extend through it to the differential which is on the same center. Two needle type roller bearings inside the cluster gear hub support the ring gear and cluster gear assembly on a hardened steel tube pressed into the axle housing. Ring gear thrust is taken by a bronze thrust plate between the end of the cluster gear hub and the axle housing. No adjustment of the stem pinion is provided but endwise adjustment of the ring gear is effected with shims between the axle tube housing and the transmission case.

Transmission countershaft is located ahead of the center line of the differential and driving wheels and is supported at each end by ball bearings housed in the case. There are two sliding gears and one idler gear on this shaft. The sliding gears are splined to the shaft and the idler gear is bushed with a gray iron bearing

Fig. 3—Exploded view of transmission gears, shafts and bearings. Differential is contained in bull-gear cluster



which turns on the shaft. The first countershaft sliding gear meshes with either the cluster gear previously described or with the reverse idler gear which is driven by the cluster gear, thus the position of the first sliding gear determines the direction of rotation of the countershaft and provides forward or backward travel of the truck.

Second countershaft sliding gear, in one position, engages the low idler gear by means of a dental clutch and makes the idler gear a driving gear. In the other position, the gear meshes with the smaller of two gears in another cluster, providing the final drive to the differential. These two positions of the second sliding gear provide two travel speeds in either direction. Both sliding gears are held in position by spring-loaded poppet balls mounted within the gear hubs. Detents are milled in the splined shaft to accept the ball poppets. The low idler gear which is in constant mesh with the final drive cluster gear, idles on the shaft when the transmission is in high gear and drives the truck in low gear when it is locked to the countershaft through the dental clutch in the second sliding gear.

Two-pinion differential, *Fig.* 4, mounted on the same center as the first drive ring gear, is conventional except that it is driven by a cluster bull gear which is pinned and shrunk to the outside of the differential case. Assembled hot, the bull gear is held in place by the differential pinion shaft until cooling is completed. The differential case is supported at each end by ball bearings, one mounted in the side of the transmission case and the other mounted at the center of the transmission case in a web provided for that purpose. The full floating axle drive shafts extend into the differential side gears through tubular extensions of the transmission case.

Carries High External Loads

Cast malleable iron is used in all of the housing parts, *Fig.* 5, except the transmission top cover which is cast gray iron. Malleable iron was selected rather than the more conventional cast iron for gear cases because all of the housing parts are heavily stressed by forces originating outside the transmission. The flywheel housing has a short, stiff transverse leaf spring clamped to the lower, outside surface. This spring, which may be seen in the worm's-eye view of the truck in *Fig.* 6, is used to return

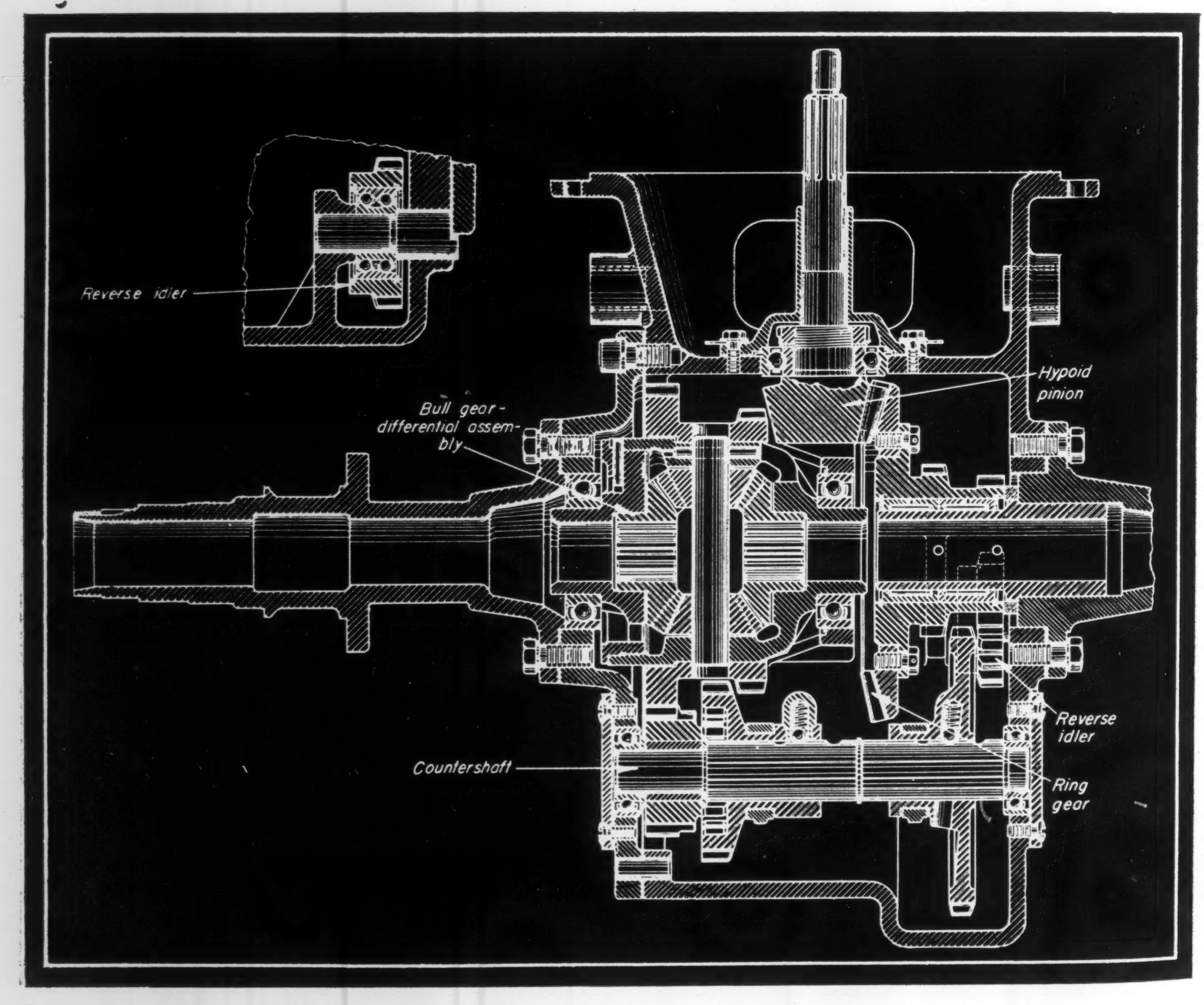


Fig. 4—Transmission assembly showing compact design of hypoid gearing, jackshaft and differential

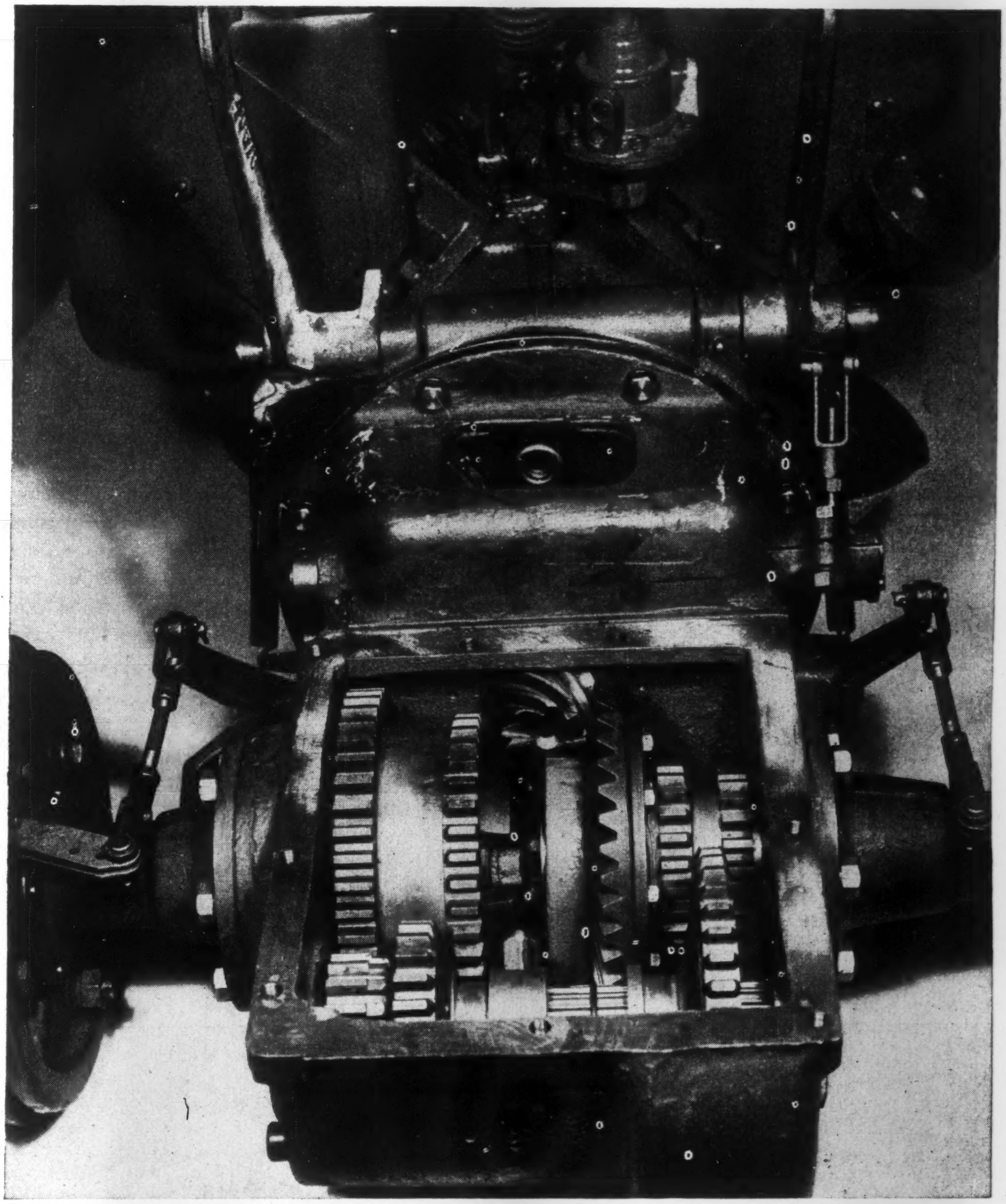


Fig. 5 — Transmission case with cover removed. Housing is malleable iron to withstand the forces originating outside the transmission. Axle drive flanges are mild steel, serving as safety shear hubs to protect internal parts

the lifting frame after it has been tilted backward with single acting hydraulic jacks. Driving and braking torques are transmitted from the center of the driving axles through the transmission case, flywheel housing and engine crankcase to the single point engine mounting at the rear of the engine.

Torsion Bar Transmits Entire Load

Bolted transversely to the bottom of the transmission case a heavy, forged steel torsion bar carries bonded rubber bushings to which the lifting frame is attached. Under maximum load conditions the entire weight of the load and the truck is transmitted to the transmission case and axle housing tubes by this torsion bar. Unusually heavy sections are used in the housing parts because added weight on the front wheels is desirable for good traction when the truck is not loaded.

The entire right side of the transmission case, *Fig.* 7, is removable for assembly purposes. This side cover is held in place by socket head capscrews and heavy dowels. The side cover is not serviced separately.

Two transmission shift levers are provided, one for forward or reverse and the other for speed change selection. The shifting forks are attached directly to these levers and engage the two sliding gears on the transmission countershaft. This two lever design eliminates the need for extra shift rails and complicated interlocking devices and hand lever travel geometry which otherwise would be essential.

Clutch and brake pedals are mounted on a dead shaft extending from drilled holes in the top side of the flywheel housing. The hydraulic brake master cylinder is bolted to the clutch housing section of the transmission case and is connected to the brake pedal by a short link. Two transverse shafts extend through the clutch housing. The upper shaft, connected to the clutch pedal by a short adjusting link actuates the clutch throwout yoke. The lower shaft actuates the parking brake levers through adjusting links and is connected to a hand brake lever mounted on the truck frame.

Conventional automotive type internal expanding hydraulic brake assemblies are bolted to flanges on the axle housings. Brake drums are cast gray iron and are integral with the wheel hubs. Divided type wheels are used with the wheel half clamping bolts arranged so that the wheel halves cannot be separated when the wheel is on the hub. This lessens the possibility of separating the wheel halves with air in the tire and makes it possible to mount wheel and inflated tire assemblies.

The combination of highest engine power and torque for a truck of this size and excellent pneumatic traction tires and heavy tire loading indicated that the propelling parts of this new truck would be subjected to extreme stresses that would not be possible in a conventional truck equipped with solid rubber tires. The axle drive flanges, splined to the axle shafts and bolted to the hubs, are relatively soft mild steel to serve as safety shear hubs to protect more expensive and difficult to service internal parts.

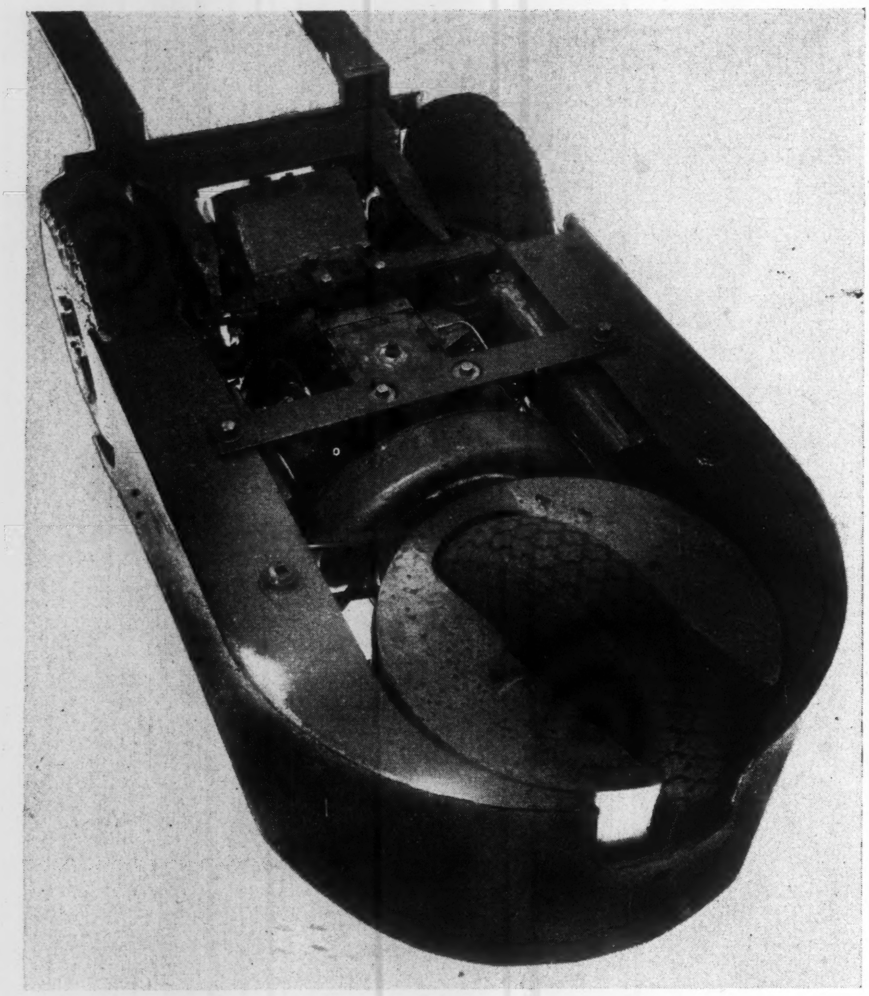
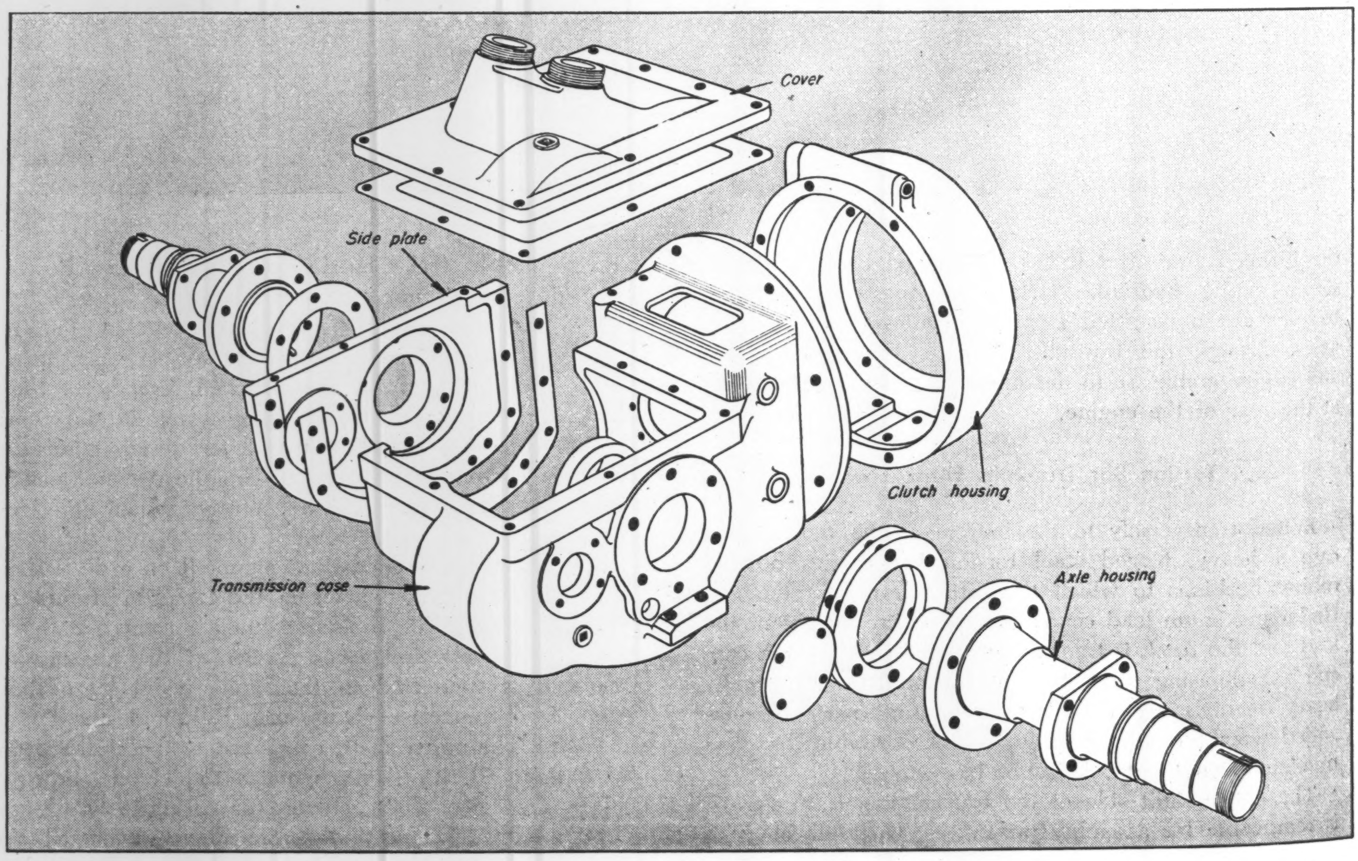


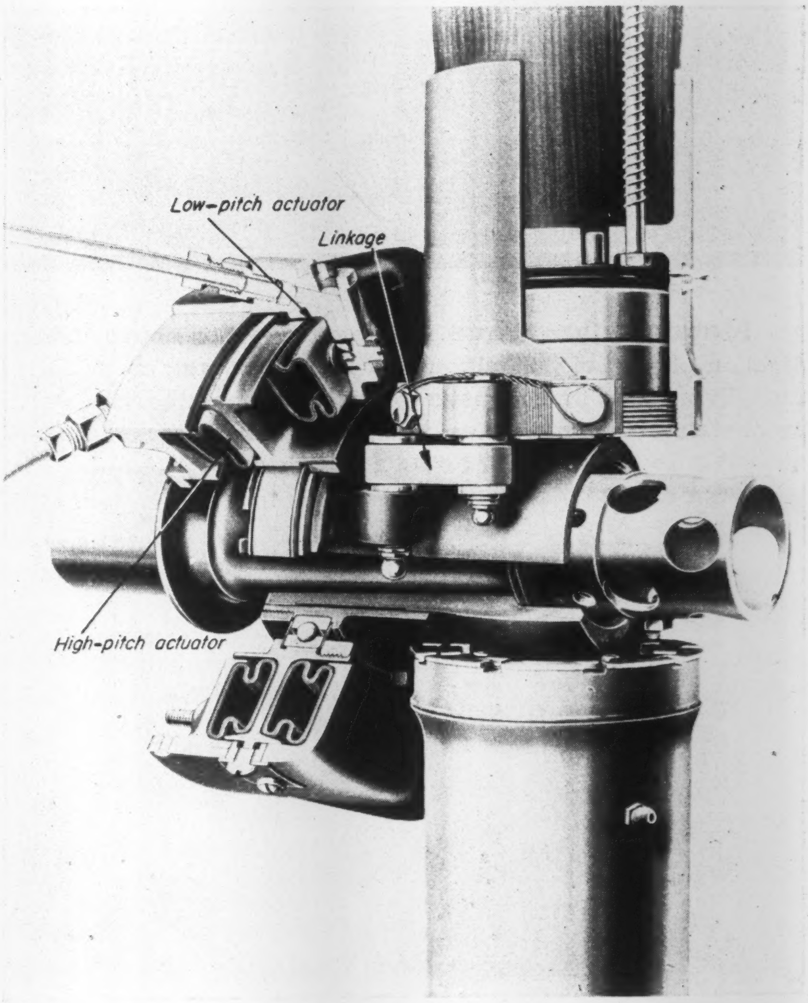
Fig. 6—Left—Worm's eye view showing power unit mountings

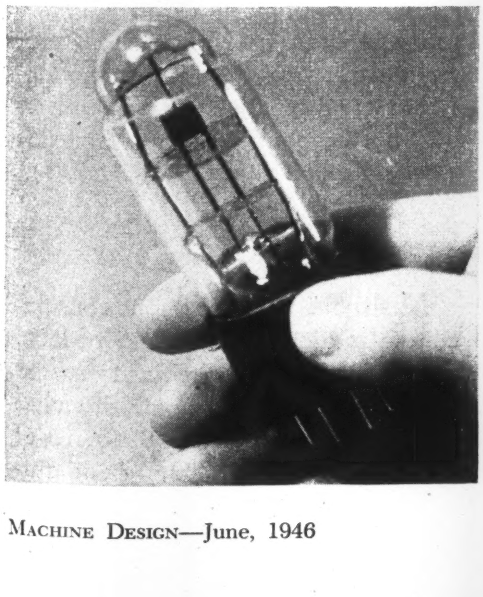
Fig. 7—Below—Exploded view of transmission housing. Entire side is removable for assembly purposes



Scanning THE FIELD for Ideas

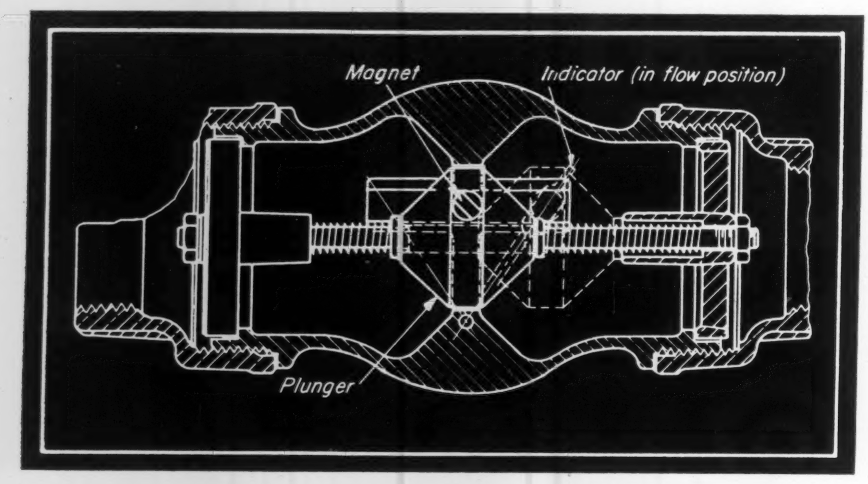
Simple hydraulic circuit of the closed system type is utilized to shift a propeller between high and low pitches as shown in the cutaway view of the actuating unit at right, designed by Continental Aviation and Engineering Corp. Propeller pitch is changed by pumping oil from one annular flexible actuator to the other. This shifts a transfer unit which is connected through a linkage to the propeller hubs. Pilot control of pitch is by a push-pull flexible cable that positions a four-way valve, reversing the flow in the circuit. Actuating power is obtained from the regular engine oil pump. Should the system fail, the propeller would automatically assume high-pitch position. The actuator unit does not revolve with the propeller; the rotary motion extends only to the thrust ball bearing in the transfer mechanism.



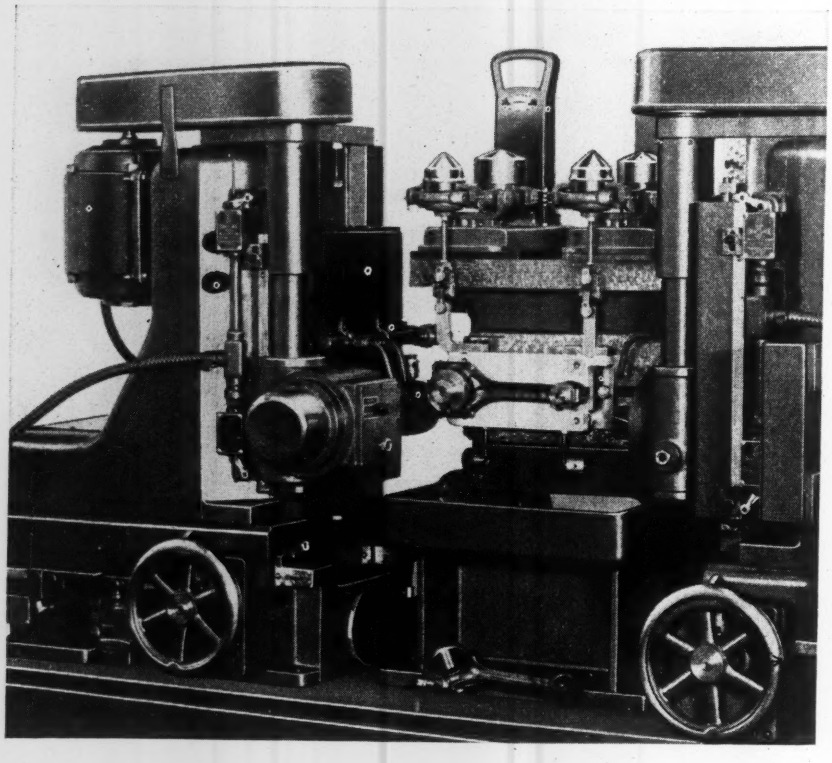


Concentrated arc lamp, having permanent electrodes sealed in a glass bulb filled with an inert gas at atmospheric pressure, is shown at left. It produces a light source so small that it may be used as a lensless projector, or with optical devices, to give from five to ten times the brilliancy upon a screen with 100 times the life of a normal tungsten projection lamp.

Source of light is a sharply defined circular spot which forms on a zirconium oxide cathode. The oxide surface is raised to its melting point by ionic bombardment, emitting a brilliant white light from this surface and a cloud of excited zirconium vapor and the inert gas which extends for a few thousandths of an inch from the cathode. This ionized vapor returns to the cathode, thus renewing the surface and resulting in lamp life of several hundred hours. Illustrated is a 10-watt lamp for operation on 21 volts direct current. It's 0.016-inch diameter light source has a brightness of 35,400 candles per square inch. Power is supplied from a special circuit involving high-voltage starting and a well-filtered rectifier or other equipment to limit current to normal values.



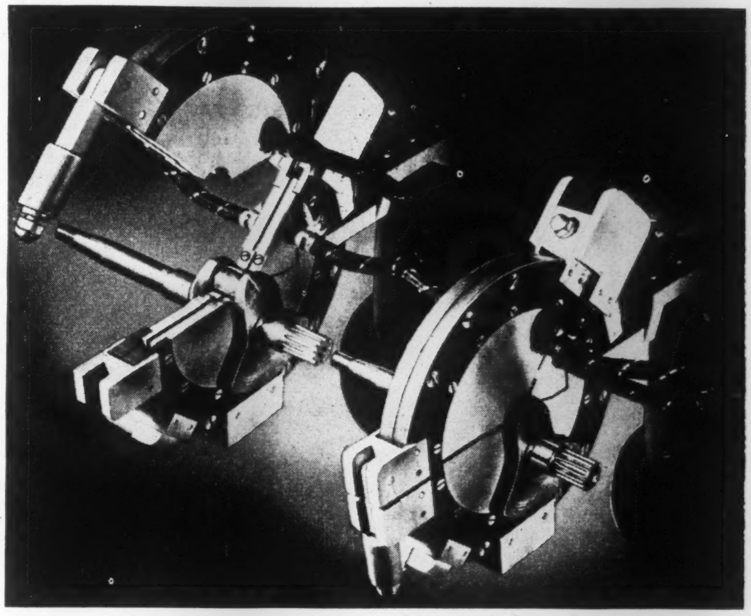
Magnetic flow gage, illustrated in section above, indicates direction of flow of fluid through a system as well as its relative rate. By utilizing the magnetic principle, seals on the indicator pin are obviated, also no pressure on the glass windows is encountered. The flow-actuated plunger is aluminum, cross bored to hold a magnetic cylinder. Springs center the plunger in no-flow position. Clearance between plunger and bore of the body when in neutral position approximates that of a loose plug so that no flow can occur without moving the plunger. Indicator hands on dials on each side of the gage synchronously follow the magnet, pointing always in the direction of flow. Inside dimensions of the gage are oversize and contoured to reduce flow restrictions to a minimum at any flow rate.



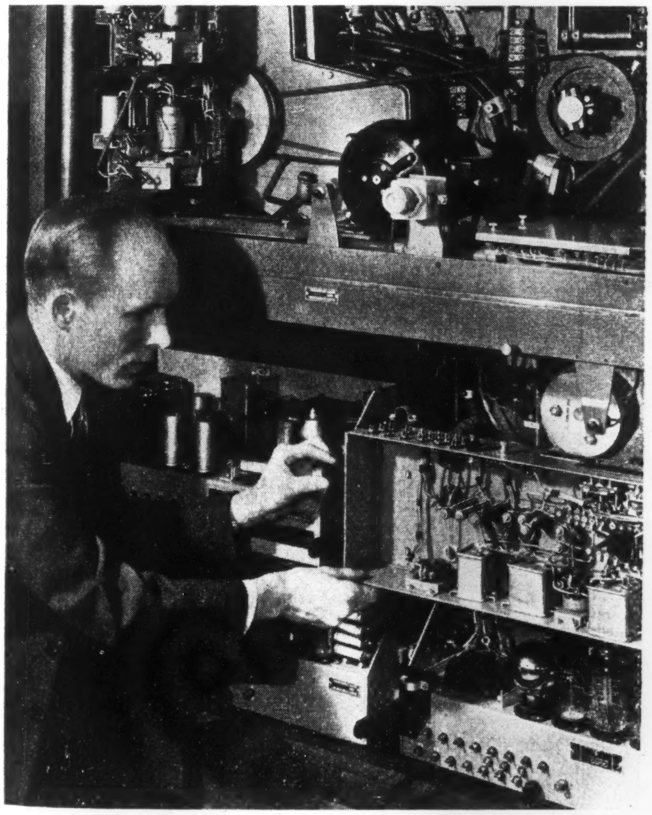
Weighing a machined part without taking it out of its fixture facilitates production, especially when parts must be of equal weight for properly balanced assemblies. To accomplish this operation in a minimum time, Snyder Tool & Engineering Co. designed and built the special-purpose machine illustrated at left below for milling connecting rods for engines. The fixture is attached to two scales which register the amount of overweight at each end of the rod, indicating the required depth of cut, which is set by positive stops. After the stops have been set the machine cycle is started, causing automatic clamping of the part in the fixture in fixed location, rapid advancing of both milling units against their stops, and feeding the cutters through the work.

Cutters then pull away to starting position, clamps release and the workpiece hangs freely on the scale rods to give an automatic check on the accuracy of the cutting operation without the necessity of removing the connecting rod from the working position.

Multiple turns in split coils facilitate induction hardening of local areas in crankshafts, camshafts and similar parts. For small diameter parts it is necessary to use coils of more than one turn to provide sufficient energy transfer. Developed by the Induction Heating Corp., the coil is made up of machined copper plates, split and hinged so that it may be clamped to provide a continuous path for current flow. As shown at right, segments are held in relationship to each other by an insulating retainer ring which runs around the outside of the coil and forms a closed passage between the coil turns. The unit also carries the quenching medium to eliminate time delay between the heating cycle and quench. Power concentrations may be produced as high as 20 kilowatts per square inch on 1/2-inch diameter shafts.

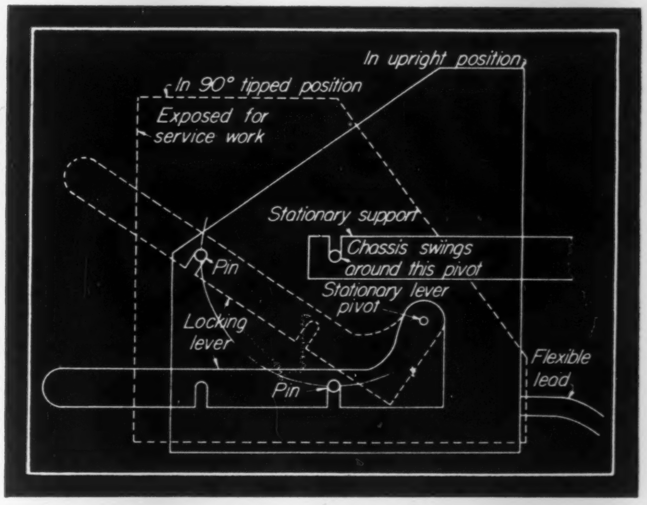


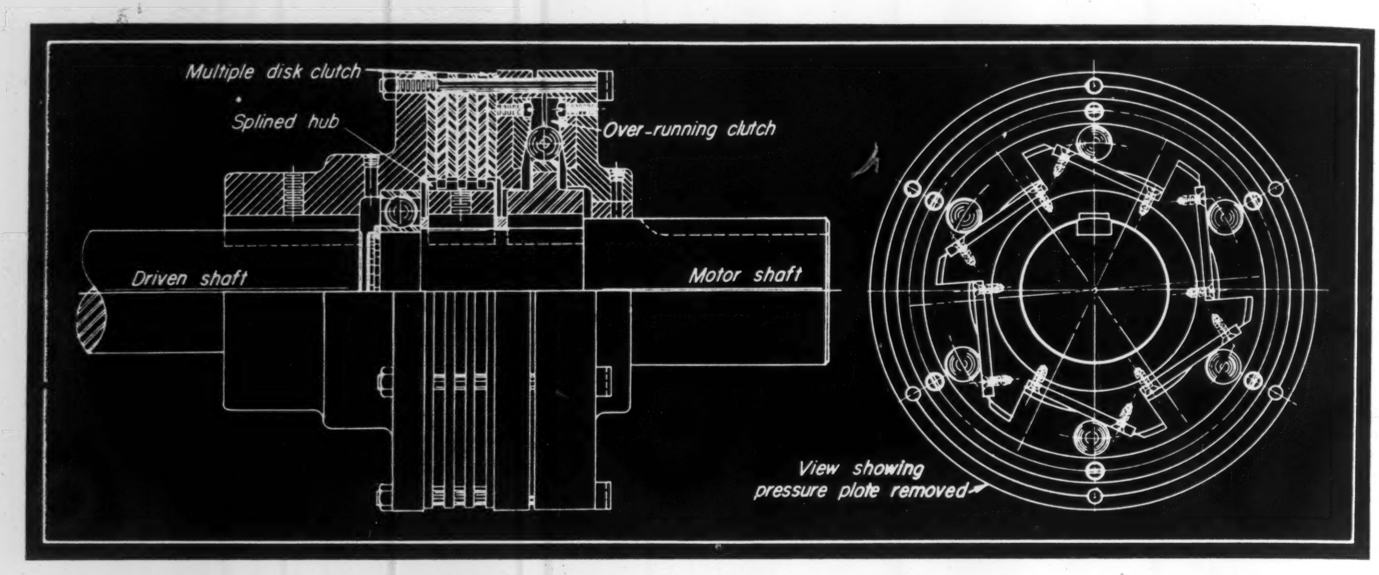
Provisions for servicing are always included in properly designed equipment, especially when obstructions or difficulties may be encountered in compactly designed units. Many an otherwise well designed machine has gained user ill will just because it was difficult to service adequately and economically. An example of how easily service may be facilitated is shown in the accompanying illustrations, above and at right, of a 35-mm projector designed by the Phillips Co. in Holland. The chassis units are all connected with flexible cables and pivoted so that they may be turned 90 degrees and locked in either position with a simple lever and pin arrangement. When the chassis is turned up, the wiring on the bottom side is exposed for service work.



Gradual pickup of load is effected in an overrunning clutch design, illustrated at the top of next page, by combining a multiple-disk clutch in the mechanism to absorb the shock of engagement. This novel adaptation also reduces the driving load on the clutch balls by transmitting the major part of the torque through the disk clutch. Only enough power is bled off through the balls to give them the required working pressure.

When the driving shaft is rotated clockwise the

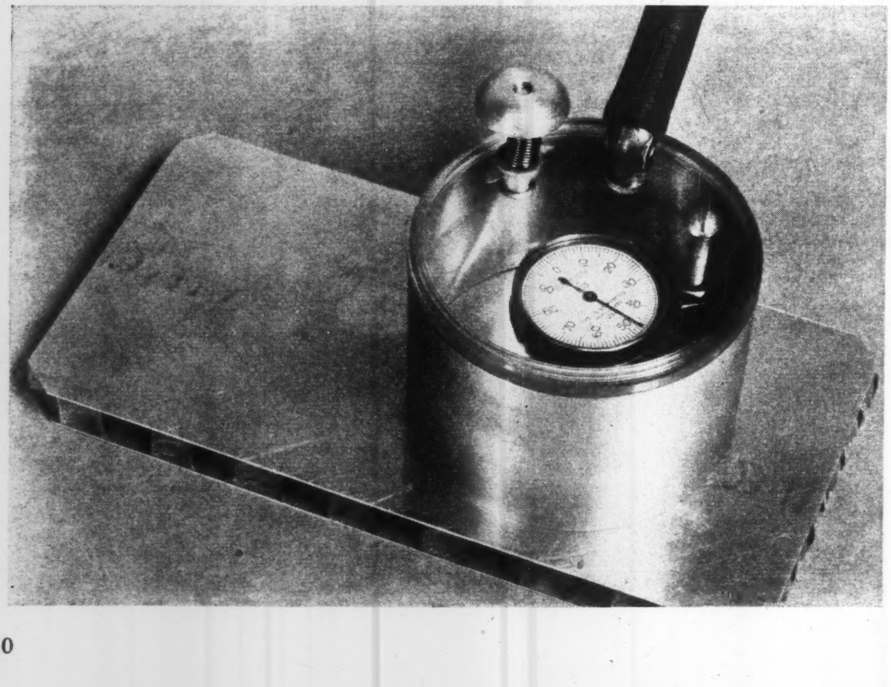




steel balls, pocketed in the ratchet-shaped drive hub, move out along hardened flat plates, receiving a radial pressure outward. During this movement the balls contact two hardened slanted surfaces and the pressure is deflected horizontally with a resultant which is a multiplication of the outward pressure. Design is such that the pressure on the balls is just enough to carry the load and is only a fraction of the ultimate pressure on the drive plates.

Power is transmitted through the splined hub and mating splined drive plates to the driven plates and thence to the driven shaft. Sintered bronze surfaces on the clutch plates are employed for their coefficient of friction and long life. Wear of the disks is automatically compensated for by increased travel of the balls during engagement. In this arrangement the balls receive a minimum of abuse by elimination of the jarring action of engagement and by driving through the disk clutch.

Measuring thickness of sheet material by placing a gage on one surface has been made possible by the development of the instrument below. Operating on the pressure principle the gage consists of an inverted cup-shaped shell with a top wall of transparent material and the bottom rim edged with a rubber gasket. Inside the shell is a dial indicator equipped with a stem reaching to a spherically shaped foot which rests against the sheet surface. Air inside the gage may be evacuated to any desired amount to give a reading, in thousandths of an inch, of the material deflection. This may be compared to the deflections of sheet of the same material but of known thicknesses. The instrument, developed by the Glenn L. Martin Co., is also useful for testing bonds in laminated materials. If the laminations are not properly cemented the amount of deflection of the gage is excessive compared to that for a properly bonded specimen, indicating a fault in the material.



Traveling collectors for obtaining power from protected conductors as discussed in the March issue on Page 110 are designed and manufactured by the Benbow Manufacturing Co. in San Francisco instead of the McCarty Co. as stated in the article. The editors are sorry for this error.

Machine Hydraulics

By Albert H. Dall
Assistant Research Director
The Cincinnati Milling Machine Co.

Part II—Balanced Circuits

IN THE previous article, on throttle-type circuits (M. D., April, 1946, Page 143), it was explained how a pressure or pressures were controlled by spring-operated valves. The controlled pressure in turn was employed to govern the flow through a throttle, thus varying the feed rate of the working slide. These spring-operated valves in every case were throttling valves in themselves, with a throttle opening which was determined by the balance between a pressure acting on the valve and a spring acting on the valve, opposing the pressure.

An ingenious device known as the balancing valve dispenses with the spring in a pressure-controlling valve. In this valve two pressures are balanced against each other. The balancing valve is applied in a modified bridge circuit, the nearest electrical analogy to which is the Wheatstone bridge, *Fig.* 12.

When applied to a bridge, the balancing valve is employed to keep the pressures at E_1 and E_2 in *Fig.* 12 mutually equal. If E_1 equals E_2 the flow through the resistances R_1 and R_2 will be inversely proportional to the respective resistances through which the flow occurs.

How the balancing valve accomplishes this objective is illustrated in *Fig.* 13. Thus if P_2 exceeds P_1 the valve will move to the left, decreasing resistance R_4 and increasing resistance R_3. This will cause P_1 to approach and finally equal P_2. The flow in each leg of the bridge is determined by the equation

$$P_0 - P_1 = q_1 R_1 \qquad (11)$$

$$P_0 - P_2 = q_2 R_2 \qquad (12)$$

Since the valve causes P_1 to equal P_2 at all times the right sides can be equated.

The ratio of the flows will therefore be

$$\frac{q_1}{q_2} = \frac{R_2}{R_1} \qquad (13)$$

It will be noted in Equation 13 that the flows q_1 and q_2 are independent of the pressures in the system. If a constant quantity Q is pumped into the junction at P_0, the flow in each will be divided in the ratio determined by Equation 13. If the resistances R_1 and R_2 are made exactly equal, the flows in each leg of the bridge will be

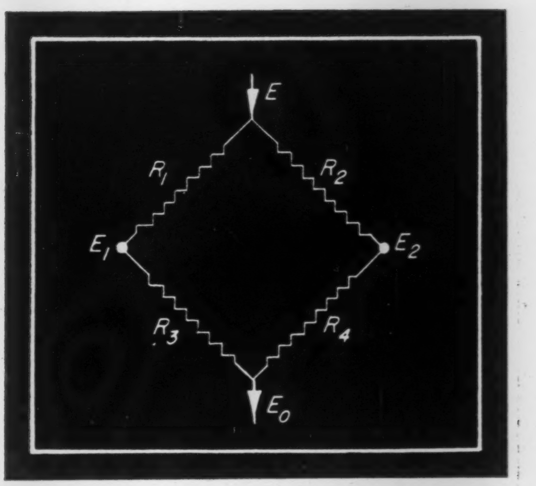


Fig. 12—Above—Wheatstone bridge circuit—the electrical analogy to the balancing valve

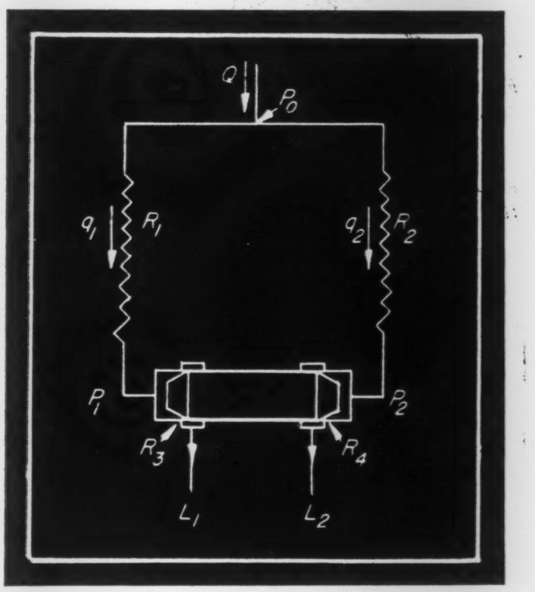
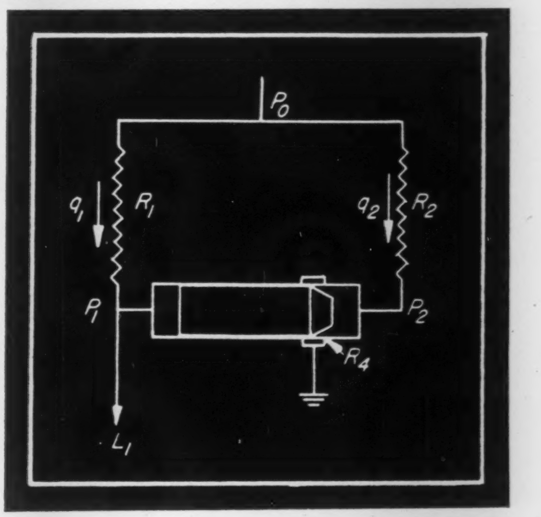


Fig. 13—Above—Balancing valve, in bridge circuit, maintains equal pressures at P_1 and P_2

Fig. 14—Below—Balancing valve in shunt circuit controls line with increased efficiency due to elimination of one variable resistance



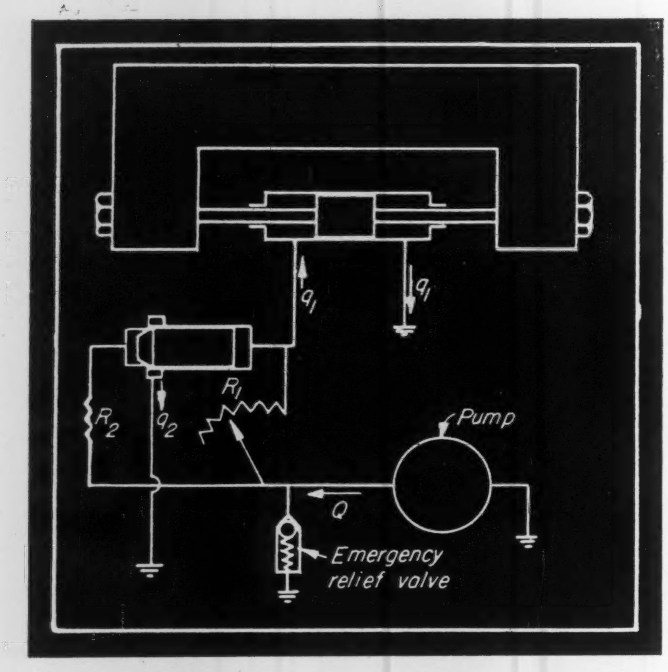
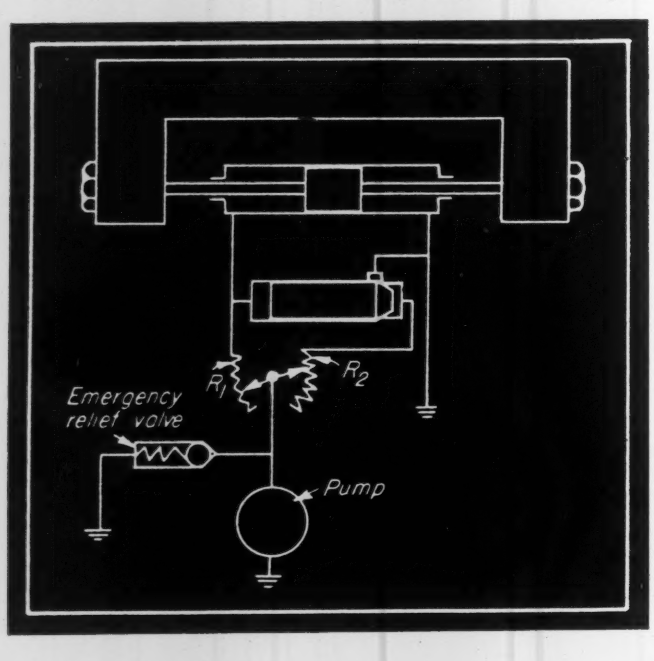


Fig. 15—Above—Application of balancing valve to a throttle circuit, with shunt through the fixed resistance R_2 and the balancing valve

Fig. 16—Below—Circuit using balancing valve and double throttle, with improved efficiency over that of Fig. 15



exactly equal irrespective of the loads L_1 and L_2, either of which can be zero.

Balancing Valve in a Shunt Circuit: In *Fig.* 14 is shown an application of the balancing valve in a shunt circuit. The automatically varying resistance, R_3, has been eliminated. The pressures P_1 and P_2 will be held equal by the action of the valve, and the flows again will distribute themselves according to Equation 13. If the line L_1 conducts the fluid to do useful work in the system, the elimination of resistance R_3 will have the effect of increasing the efficiency of the system.

In *Fig.* 15 is shown an application of the balancing valve to a throttle circuit. In this circuit the shunt is through R_2 and through the balancing valve to the sump. The feed rate is determined by the ratio of the fixed resistance to the sum of the variable and fixed resistance:

$$q_1 = Q\frac{R_2}{R_1+R_2} \quad \text{.......} \quad (14)$$

The circuit in *Fig.* 16 is identical with that of *Fig.* 15 with the exception that both R_1 and R_2 are varied in a complementary manner. With this arrangement higher efficiencies are obtained because the energy lost in the parallel resistances remains more nearly constant with change in feed rate. Thus when a large quantity of fluid is passing through the shunt its resistance will be low and when a small quantity is passing through the shunt its resistance will be high.

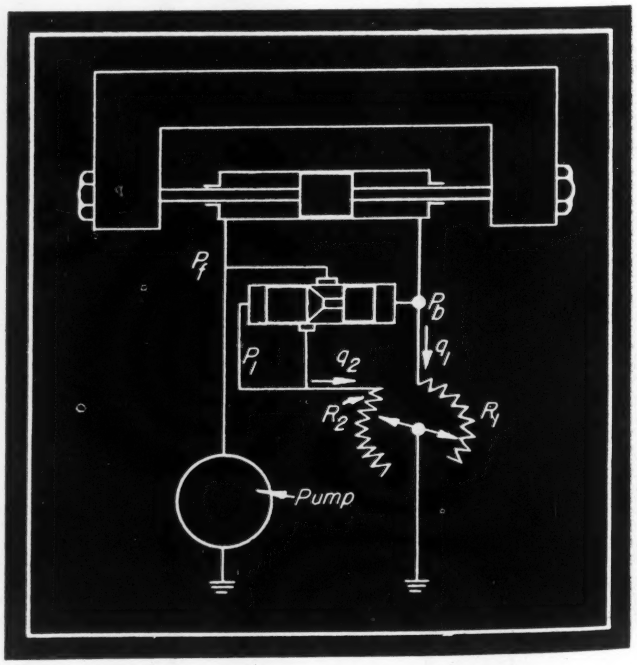
Balancing Circuit with Back Pressure: Where a back pressure on the working cylinder is desired, the circuit shown in *Fig.* 17 will control the feed rate by balancing the back pressure against the shunt pressure. The pressure P_1 will be kept equal to P_b by the action of the valve. If the throttle port on the balancing valve is wide open, P_1 will equal P_f. Since P_f is always higher than P_b, the valve will move to the right to close the throttling port, causing P_1 to diminish in value. Balance will be reached when P_1 equals P_b.

Since the pressure drop across both resistances is identical, the flows q_1 and q_2 will respond to Equation 13. It is obvious that this system will lose control when the back pressure exceeds the forward pressure, as in the case of a negative work load.

Balance Circuit for Negative Work Loads: In *Fig.* 18 is shown an arrangement for the application of a balancing valve in a circuit where the work load is always negative. This circuit will operate correctly when and only when the forward pressure is less than the back pressure.

In the figure, the forward pressure P_f is balanced against

Fig. 17—Circuit with balancing valve which balances back pressure and shunt pressure



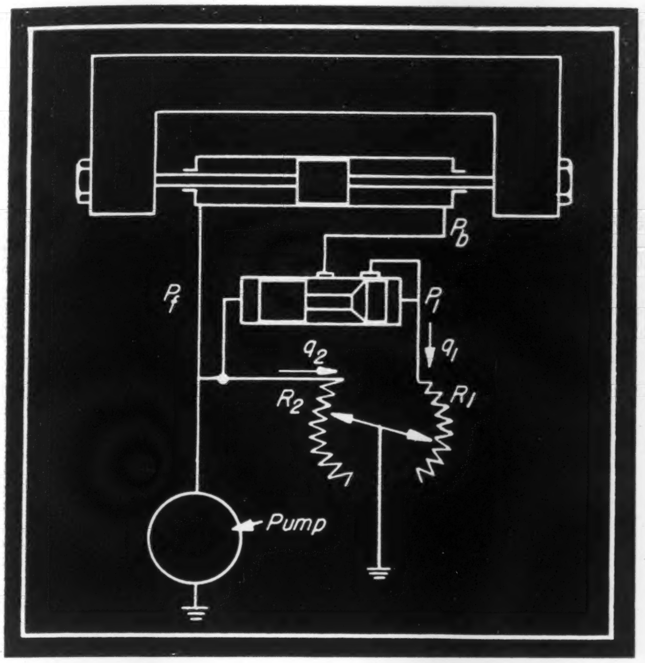


Fig. 18—Above—Application of a balancing valve in a circuit where the work load is always negative

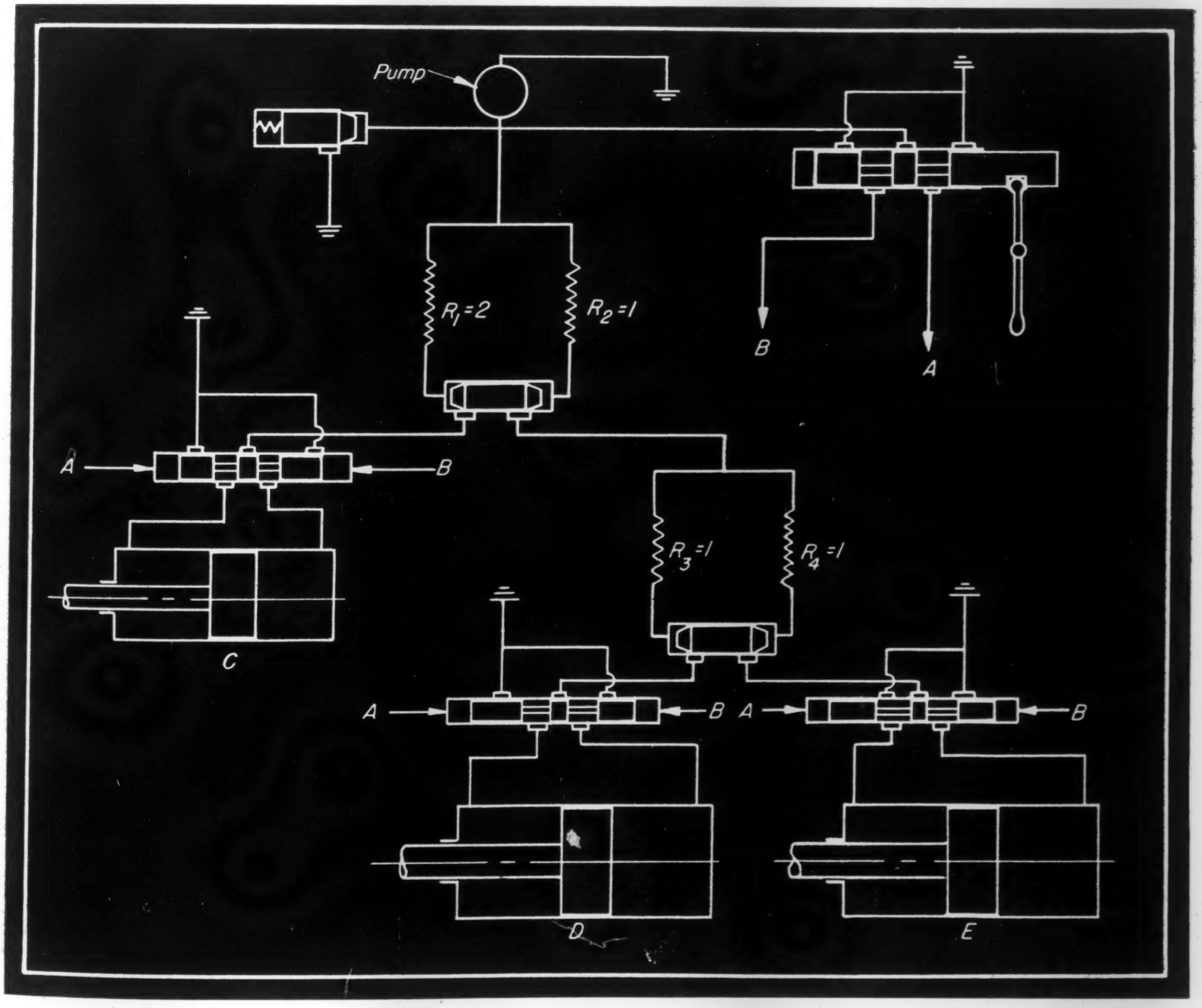
the pressure P_1, which is the pressure before the throttle. Thus the ratio of flow through the shunt R_2 to the flow through the throttle will be in inverse proportion to their respective resistances, as shown by Equation 13.

Forward and back pressures on the motor are determined by the loads. This would be true even if the balancing valve were eliminated and the circuit contained only a shunt resistance and a resistance in the back-pressure line. The function of the balancing valve is to control the rates irrespective of load on the system. The criterion which determines whether or not the balancing valve operates to cause P_f to equal P_1, is whether or not the valve moves correctly when one of these pressures is higher than the other. Thus, in *Fig.* 18, if P_1 is less than P_f the valve will move to the right, opening the throttle in the valve and causing P_1 to approach P_b, which is higher than P_f. The converse will occur when P_f is smaller than P_1.

It is obvious that the circuits of *Fig.* 17 and *Fig.* 18 can be combined to form a circuit which will control the rate with either positive or negative work resistance.

Control of Multiple Pistons: The balancing valve can be used for controlling the rate of a single slide as

Fig. 19—Below—Balanced circuit for three pistons all of which move at same rate regardless of load differences



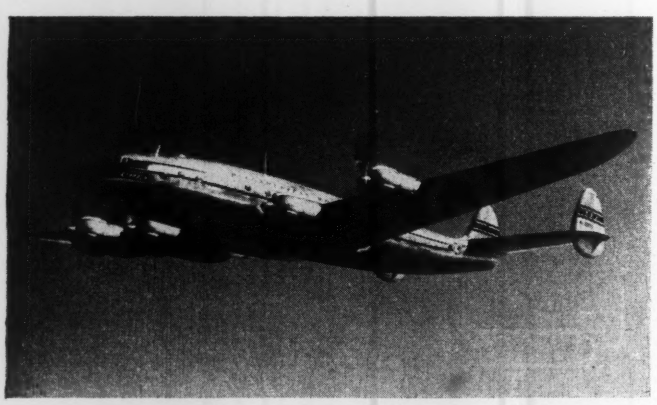
shown in the various circuits. It also can be used to control the rates of more than one slide simultaneously. The rate of the multiple slides can be held in exact relation to each other by means of the proper resistance relation.

In *Fig.* 19 is shown a system of three equal area pistons, all of which will move at the same rates irrespective of load, if the ratio of resistances shown are used. The resistance R_1 is double that of R_2. Thus, from Equation 13, exactly twice the flow will occur through R_2 as through R_1. The flow through R_2 is then divided again equally. All the flows will be equal, regardless of the load encountered on any of the pistons, up to the point where the relief valve opens. The unique nature of this performance will be realized when it is considered that if one of the pistons encounters an obstruction, all of the pistons will stop. For example, if piston C meets an obstruction, the pressure on the left of the first balancing valve will reach pump pressure which, in turn, will stop all flow to D and E.

In conclusion, it may be stated that the balancing valve performs a very unusual function in hydraulics; namely, the control of flows without respect to pressure levels or loads on the system. A subsequent article will deal with the design of hydraulic circuits to utilize the unique characteristics of variable-delivery pumps.

Here's the Constellation

FIRST of the pressurized 350-mph transports to enter airline service, Lockheed's new Constellation has ocean-spanning range and power to lift more than 22 tons of useful load. Heated walls and floors warm the pressurized cabin uniformly without drafts or cold spots, and automatic air conditioning provides fresh air on the ground or at altitude. Latest insulating methods effectively control flight noise and vibration, engine vibration being damped out by an ingenious system of floating engine mounts.



For the first time in a four-engine plane, pilots have easy positive control in every flight condition by application of hydraulic power boost to all control surfaces. The pilot's flight controls also connect directly to the control surfaces so that there is no loss of "feel". Air for cabin pressurization is furnished by dual superchargers direct-driven by the engines, either supercharger being able to take over the entire load in case an engine should fail.

Four Wright Cyclone 18-cylinder, air-cooled engines wth a total output of 10,000 hp place the Constellation almost in the pursuit plane class. Any two of the four engines permit the fully loaded plane to fly normally and even to climb to 8000 feet at maximum gross weights.

Leading Particulars of the Constellation

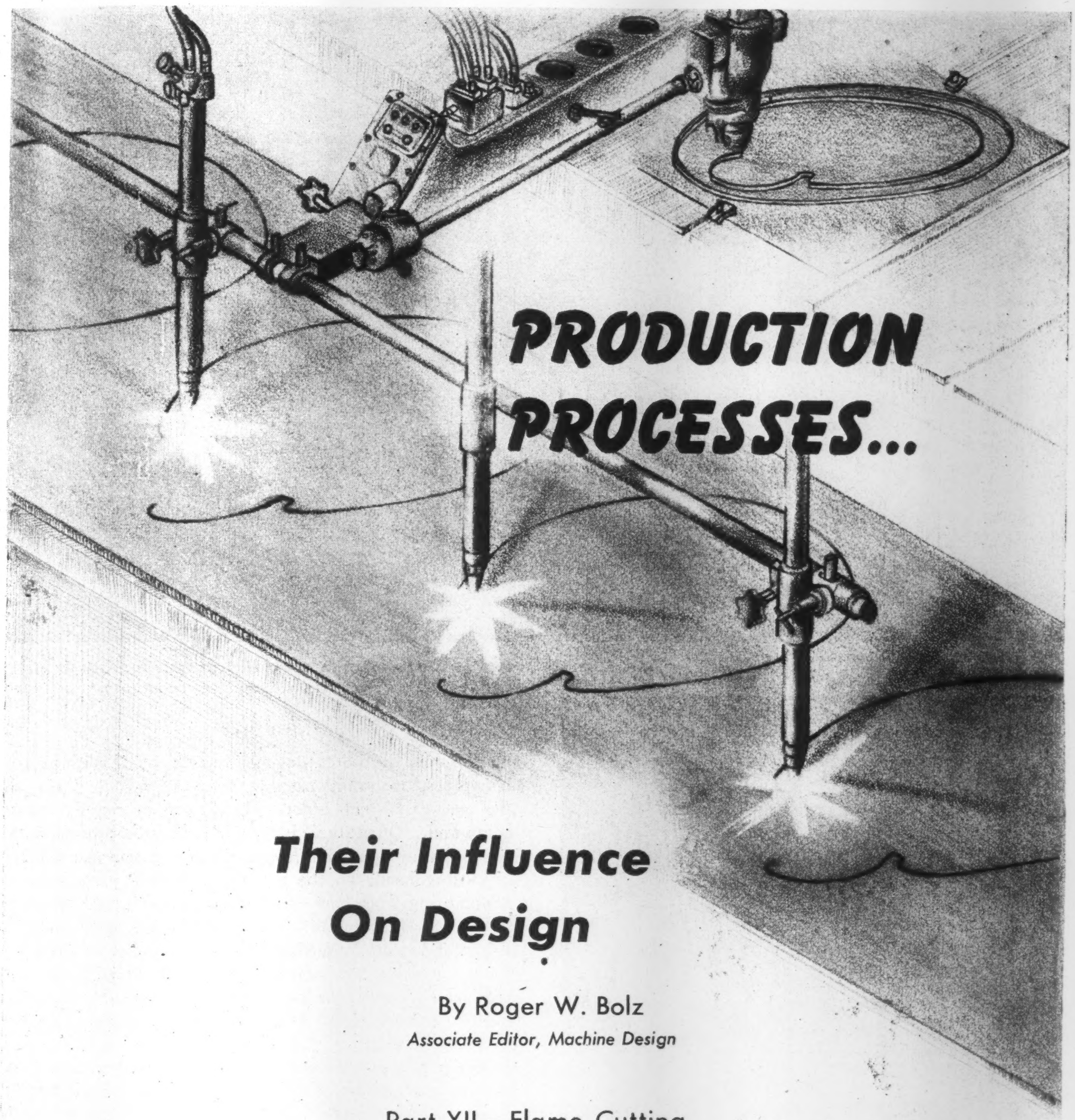
Span	123 ft 0 in.
Overall length	95 ft 1 in.
Height	23 ft 8 in.
Wing area	1650 sq ft
Maximum gross weight	100,000 lb
Payload	20,000 lb
Maximum speed at 20,000 ft	350 mph
Cruising speed at 20,000 ft	300 mph
Stalling speed at sea level	80 mph
Fuel consumption	approx. 1 mpg
Range, at cruising speed	approx. 5000 mi
Service ceiling	in excess of 25,000 ft
Take-off distance over 50-ft obstruction	2800 ft
Landing distance over 50-ft obstruction	2500 ft
Passengers, seated	48 - 64
Passengers, in sleeping berths	26
Crew	6 - 7

Aerodynamic lines reduce drag to a minimum. The wing is a development of that used on the Lockheed P-38 Lightning fighter. It embodies the same high-lift and low-drag features with the P-38's stability and control in flight and landing. The fuselage has a perfect circle cross section to permit easy sealing for pressurization. Its shape follows the lines of an airfoil with a slight fore and aft camber that provides maximum pilot visibility and minimum nose landing-gear weight. The tricycle landing gear with dual wheels and a steerable nose gear has dual hydraulic braking for easy ground maneuvering.

Developing Atomic Energy Controls

CHIEF obstacle in the way of developing atomic power will be the difficulty of organizing a large-scale industrial development in an internationally safe way. This presents, actually, problems much more difficult to solve than any of the technical developments that are necessary. It will require an unusual amount of statesmanship to balance properly the necessity of allaying the international suspicion that arises from withholding technical secrets, against the obvious danger of dumping details of the procedures for an extremely dangerous new method of warfare on a world that may not yet be prepared to renounce war. Furthermore, the proper balance should be found in the relatively short time that will elapse before the "secrets" will naturally become open knowledge by rediscovery on the part of the scientists and engineers of other countries.

One might be led to question whether the scientists acted wisely in presenting the statesmen of the world with this appalling new problem. Actualy there was no choice. Once basic knoweldge is acquired, any attempt at preventing its fruition would be as futile as hoping to stop the earth from revolving around the sun by decree.—*From a paper by Dr. Enrico Fermi presented at the recent George Westinghouse Centennial Forum in Pittsburgh.*



PRODUCTION PROCESSES...

Their Influence On Design

By Roger W. Bolz

Associate Editor, Machine Design

Part XII—Flame Cutting

EVOLUTION and development of flame cutting—the severing of ferrous metals by rapid oxidation from a jet of pure oxygen directed at a portion heated to the fusion point—has been surprisingly rapid in the short period since its introduction. Wide acceptance and utilization of all-welded machines and machine parts have brought into sharp focus the mass production of component units by means of accurately controlled mechanical flame cutting machines. For the machine designer this development eclipses the many other uses of gas cutting such as manual cutting, scarfing, gouging, descaling, lancing or piercing and cutoff.

Today, mechanical flame-cutting machines take their place alongside other production machines in industry and have opened a vast field for quantity manufacture of many locomotive, crane, ship, engine, and other parts hitherto restricted to somewhat slower methods, *Fig.* 1.

Size of parts which can be produced is limited primarily, if not entirely, by the size of the table for the machine on hand. Likewise, metal thickness is

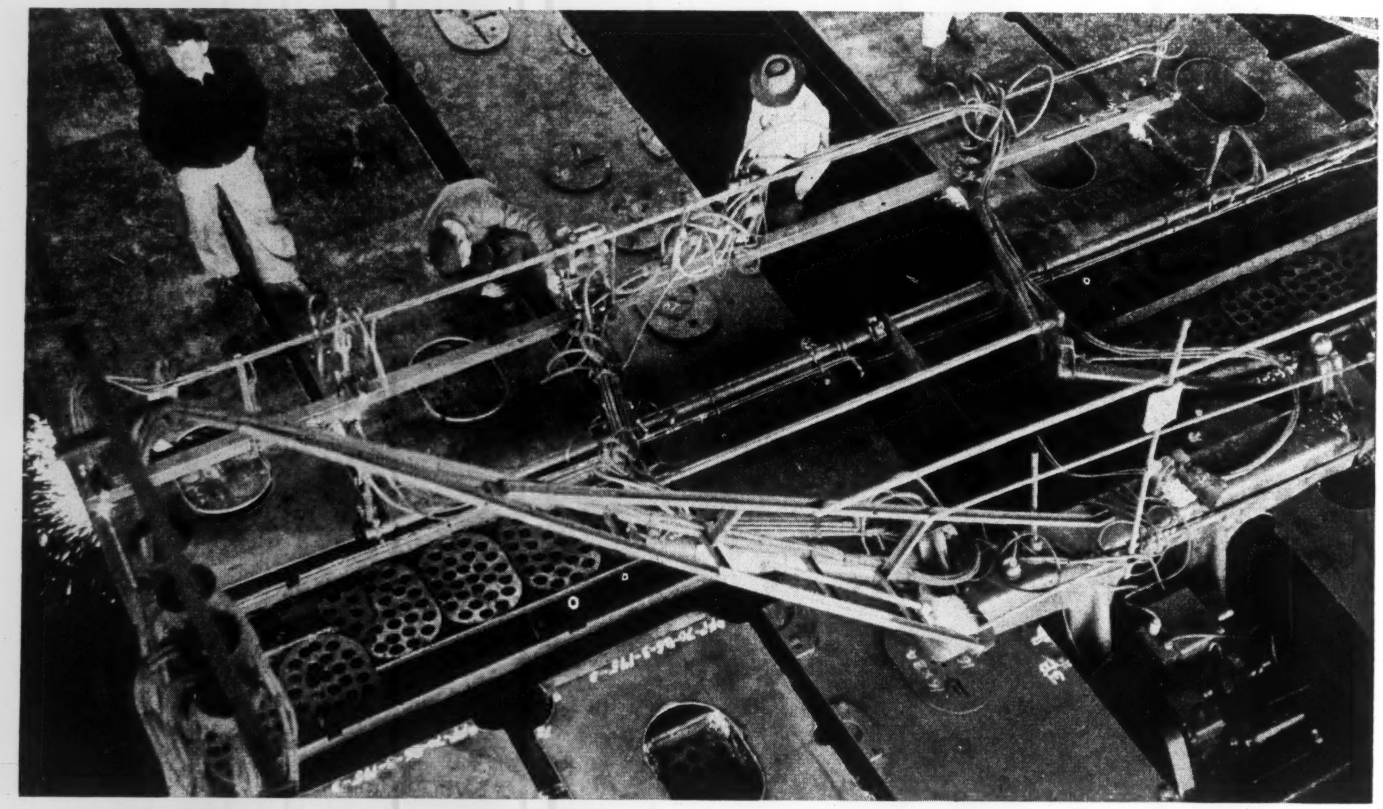
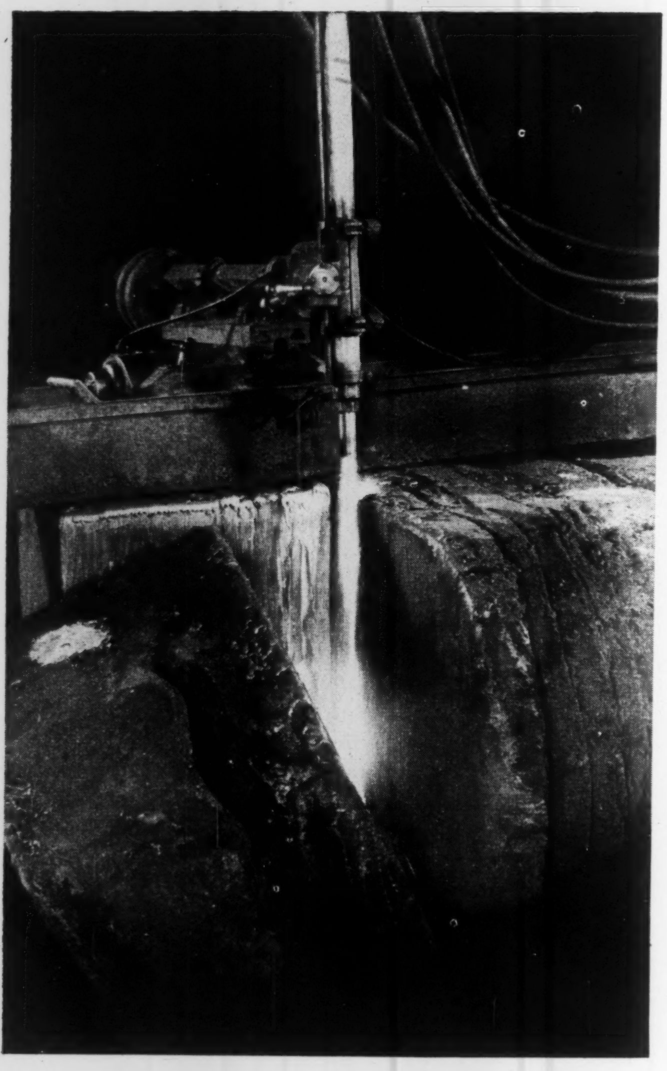


Fig. 1—Flame shaping solid floor panels for Liberty ships with a Travograph type traveling pantograph machine



limited only by the cutting heads available. With proper blowpipes, materials ranging from light-gage sheet all the way to 60-inch billets, *Fig.* 2, can be readily cut or contoured. Quantity of parts which may be produced depends mainly upon the design of the part and the number of cutting heads on the machine. Simple parts such as annular rings and simple plain shapes may be cut from plate in considerable quantities by means of the stack method, *Fig.* 3, if a multiple-head machine is available. With a six-torch machine cutting stacks of six ¼-inch plates to a master template, records from one particular shop show that in a period of 15 hours and 50 minutes including setup and cutting time, a total of 2724 pieces were cut at a rate of 172 pieces per hour. On the other hand, however, a part of complicated design which necessarily must be cut from a heavy billet naturally will limit severely the quantity of production. Linear cutting speeds with machine gas cutting are indicated generally in TABLE I. These speeds, though, are subject to modification by the existing operating conditions, material, machine, intricacy of contours, cutability of the material, etc., and may run from around 1 to 1.5 inches per minute with the heaviest sections to as high as 75 inches per minute—a speed which has been used successfully on certain stationary shape-cutting machines.

With production in any quantity whatever, where ferrous materials to be cut are bulky or have heavy sections to be removed, the flame-cutting process offers many advantages. Relatively little power is required to operate the mechanisms of a mechanical flame-cutting unit and to re-

Fig. 2—Left—An Oxweld CM-21 cutting machine with a blowpipe capable of cutting material 60 inches in thickness

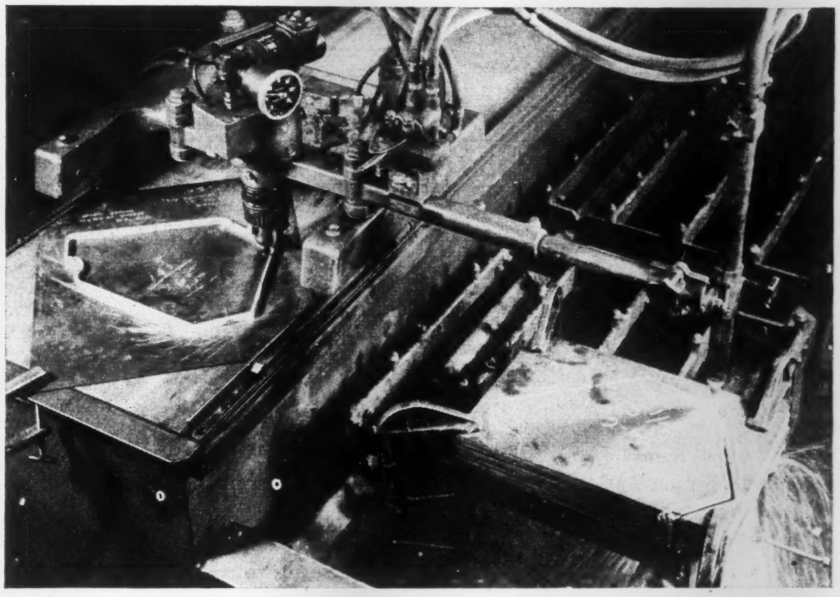


Fig. 3—Right—Stack cutting with an Oxweld CM-15 machine. Some 14 pieces are being cut from 3/16 inch plates

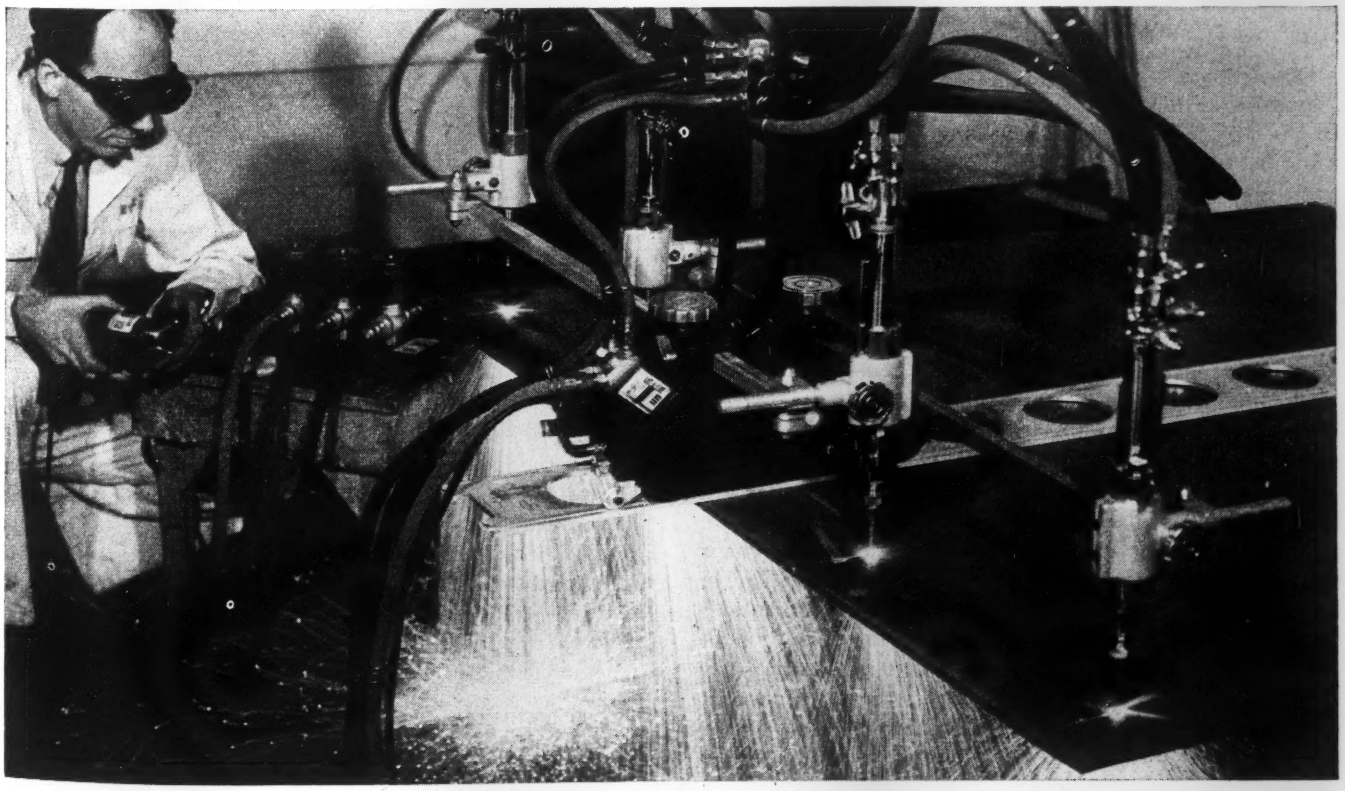
move or cut the metal. Cost is made up primarily of the cost of the gases used in cutting. Initial investment and overall operating cost of gas-cutting equipment, as compared with purely mechanical methods for cutting ferrous metals, are considerably lower, and speed of metal removal per unit volume is far greater.

Certain limiting factors or undesirable characteristics attendant to the heating must be considered. The heat created during cutting tends to relieve or redistribute the lock-up stresses in steel plate or billets in addition to causing general expansion. This results in warpage or distortion in the finished part which is especially undesirable where highest accuracy is desired. No specific rules can be set forth disposing of this matter and each case must be considered and worked out separately. A number of methods now in use for obviating or compensating for distortion are: (1) Adjustment of template size; (2) use of dual-blowpipe operation, either 180 degrees apart on symmetrical shapes or operating in parallel on rectangular-type work; (3) clamping or wedging; (4) cutting at elevated temperatures; and (5) cutting under water.

Cutting machines available today might be divided into two general groups—portable and stationary types. The portable machines, which are usually limited to operations on material up to 10 inches thick, can be further modified into straight-line cutting machines, cutoff machines and shape-cutting machines.

Flexibility is probably the most outstanding advantage in portable-machine cutting. A number of economies result from the fact that the machine can be brought to the work rather than the work brought to the machine. Investment in equipment is lower than with other methods and specialized work can be handled as readily as production work. As a rule, portable straight-line machines, *Fig.* 4, are usually recommended for use only to assure what is termed "structural accuracies". That is, for steel-structure fabrication such as required in shipbuilding, lo-

Fig. 4—Below—Portable straight-line cutting machine making four cuts simultaneously



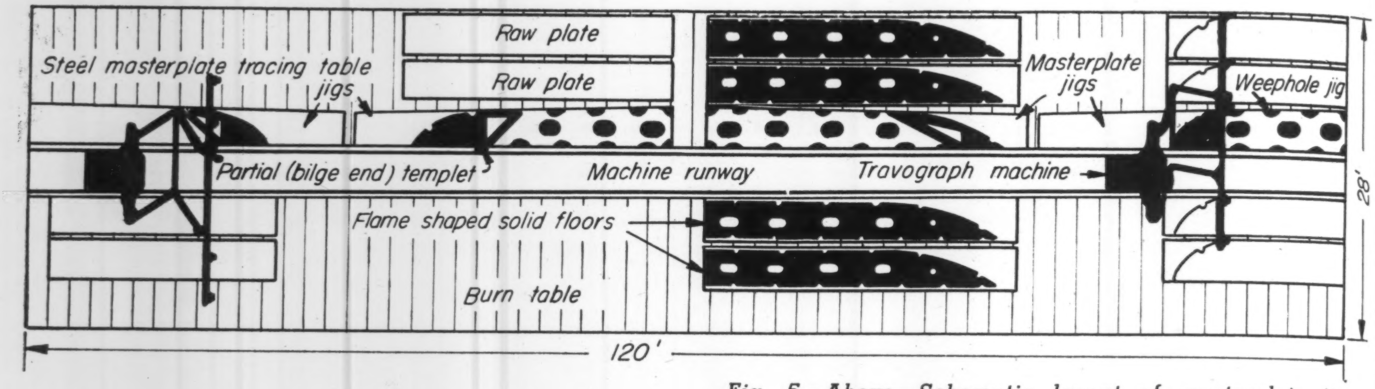


Fig. 5—Above—Schematic layout of masterplate jigs, burn tables and runway of Travograph used in producing parts for Liberty ships

comotive work and similar machine parts where welding type edge preparation is of sufficient accuracy. A limited amount of contour cutting for plate-edge preparation can be done with this type machine.

In general, stationary machines, i.e., those requiring movement of material to the machine, includes: Straight-line, shape-cutting, production cutoff, flame planers, surfacers, and special-purpose machines. The straight-line machines are similar to the portable type but are more sturdily built, are heavier, carry a greater load, and perform to more exacting accuracies. This is because a more rigid and carefully aligned track can be used.

Similarly, stationary shape-cutting machines in a wide variety are available with hand-operated, mechanical, magnetic, or electronic tracing devices. Multiple blowpipes can be used on these machines and the longitudinal working area can be increased indefinitely by extensions to the permanent track and the cutting and tracing tables. Utilization of these possibilities assisted greatly in the production of wartime Liberty ships, *Fig.* 5. These machines are used with as many as six blowpipes to further production. In addition, stack cutting, *Fig.* 3, can be used.

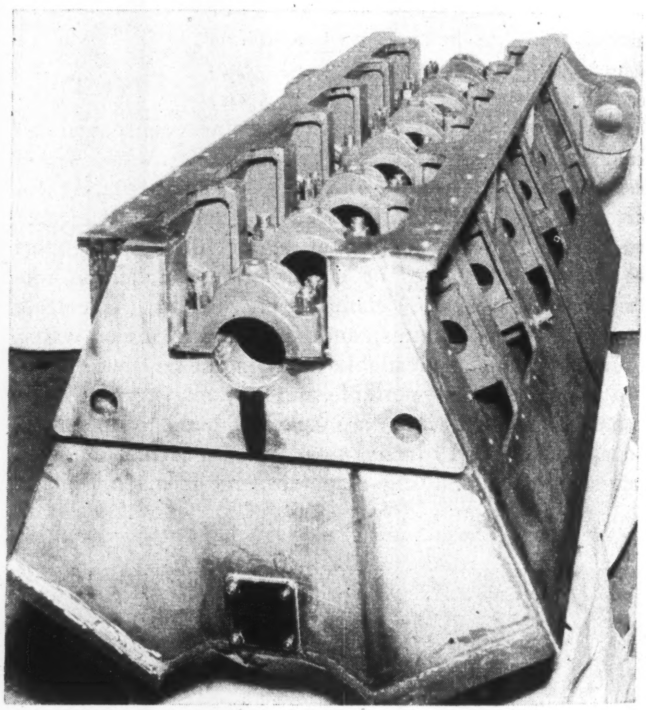
Aids Fast, Automatic Welding

Development of fast, automatic welding led to the development of the so-called flame planer type of machine for the rapid production of long, straight parts. The flame-cut edge produced is sufficiently accurate that further preparation is unnecessary. Three of these machines can operate on four sides of a plate simultaneously and triple-face cuts may be produced at one pass.

Many special-purpose machines are also in use, mostly for processing raw material rather than finished parts, consequently these do not merit attention at this time.

Designs: Generally speaking, the design possibilities available by means of flame cutting are legion as has been exemplified by the multitude of all-welded machines produced in recent years. The inability to project a jet of flame and oxygen in other than a straight line imposes a limitation on design similar to that in contour sawing, since curvilinear contour cutting is thus impossible. Perhaps the closest approach toward a solution of this problem to date has been the several multiface edge preparations such as double bevel to shoulder, and the formation of continuous J-grooves with the use of gouging nozzles. In gouging, the oxygen jet impinges upon the plate edge in such a way that the deflection of the jet results in a continuous curved surface. Applications of this phase of the process at present are limited to the preparation of plate edges for welding.

Fig. 6—Below—All-welded 400-hp, 12-cylinder gasoline engine crankcase with flame-cut components



Changes in the design of parts produced by flame cutting can be made rapidly and at little expense. Templates for magnetic tracer machines or drawings for electronic tracer machines can be replaced or revised easily. In many cases, savings in weight are obtained by substituting steel fabrication for cast design, *Fig.* 6.

Introduction of electronically controlled tracing mechanisms has made possible extremely accurate reproduction of intricate designs, *Fig.* 7. These machines extend the possibilities of the process, which formerly were restricted somewhat by the inability of the follower wheel of magnetically controlled machines to track the most intricate details without the use of elaborate jigs and some additional machining of the cut piece. However, the sprocket wheels shown in *Fig.* 8 illustrate excellently the capabilities of magnetic type machines with proper templates. The disadvantages of the magnetic tracing mechanism are most evident on internal and external corners where expensive templates are necessary to reproduce accurate small or square corners. Templates most easily produced are cut with inside corner radii somewhat larger than the radius of the magnetic follower roller to avoid locking of the roller and, as a result, invariably produce rounded corners on the cut parts. A 5/16-inch diameter roller requires a 7/32-inch minimum template radius and a 1/2-inch roller an inside template radius of 3/8-inch, minimum, thus limiting the radii on the parts to be cut. It is good practice to design parts to have radii of 1/2-inch or over wherever possible. Templates or drawings used with the electronic type tracer can be corrected to achieve exacting results with square or unusual corners.

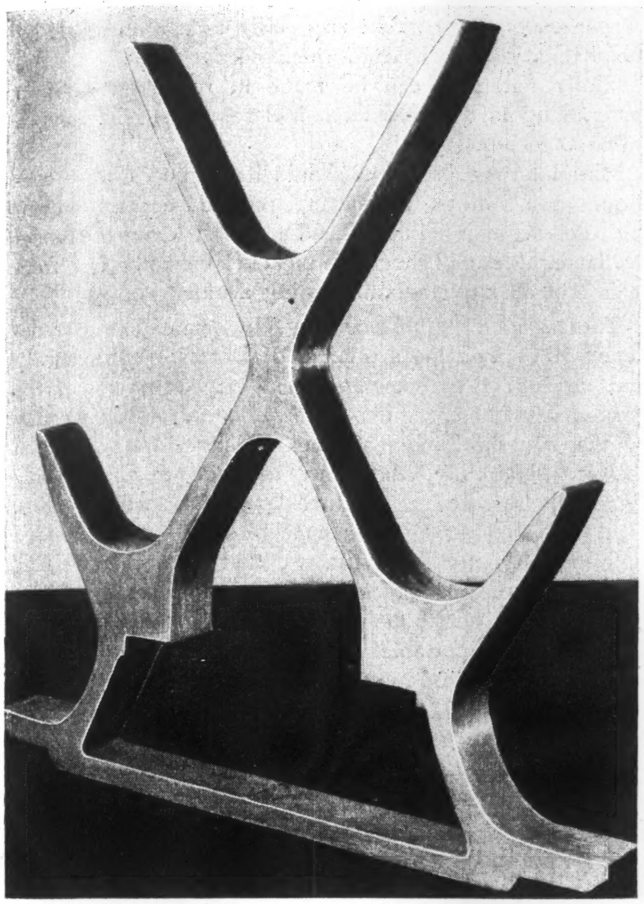
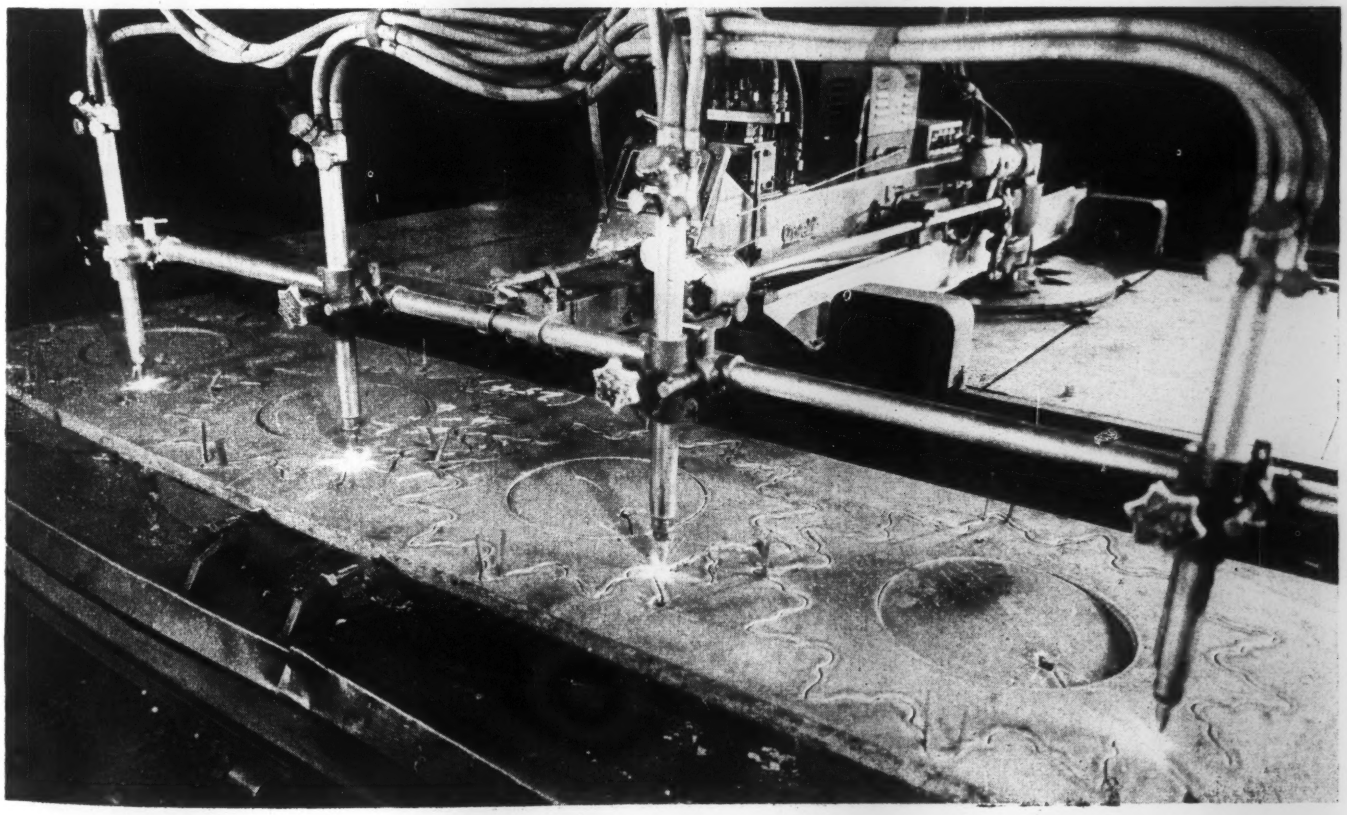


Fig. 7—Above—Machine component flame cut from hot rolled plate showing possible design intricacies

Range of Applications Wide

In designing to take advantage of low-cost raw hot or cold-rolled plate, forged slabs, rolled or drawn bar, etc., a great range of types of parts and sizes of machines can be considered within the scope of the economies afforded by flame cutting. Extremely small parts, including 1.125-inch thick bodies of 9-mm Browning pistols as well as parts and 2.031-inch thick bodies for Bren guns, were flame cut from 3.5 per cent nickel gun steel bar stock in great quantities during World War II. In direct contrast is the cutting of two crankshafts from one forged slab 25 feet long, 10 feet wide and 25 to 28 inches in thickness with a 38 to 42-inch thick end flange. One flame-cut blank crankshaft cut from this blank after preheating to 400 F is shown in *Fig.* 9. Contours are clearly shown and cutting was carried out at about 2 to 2.5 inches per minute. Following cutting, the blank is annealed and then the bearings between throws rough turned. After heating, the various throws are twisted to proper position. A final heat treat-

Fig. 8—Below—Chain sprockets being cut from a steel slab with a multiple-torch magnetic follower machine



ment is followed by completely finish machining all over.

Materials: Essentially all ferrous metals and alloys can be flame cut, although the low and medium-carbon types can be handled most easily and economically. Steels under 0.30 per cent carbon can be cut cold—without heat treatment before or after cutting. Steels in the higher alloy groups can also be cut with relative ease but require preheating before and annealing afterward.

Table I

Machine Gas Cutting

Metal Thickness		Drill Size Jet	Approximate Kerf Width	Cutting Speed Inches/minute
Min.	Max.			
1/64	1/4	72	1/32 - 1/16	17 - 35
1/4	1/2	65	3/64 - 5/64	17 - 25
1/4	1	58	1/16 - 3/32	17 - 20
1/4	2	54	5/64 - 7/64	14 - 17
1/2	3	52	3/32 - 1/8	11 - 15
2	5	50	7/64 - 9/64	7 - 14
4	6	45	1/8 - 11/64	6.5 - 9
6	10	40	9/64 - 3/16	3.8 - 6.5
10	12	35	5/32 - 7/32	3.5 - 3.8
12	14	28	3/16 - 1/4	3.2 - 3.5
14	15	20	1/4 - 5/16	3.1 - 3.2
14	16	11/64	5/16 - 3/8	3.0 - 3.2

NOTE: Lowest speeds are for maximum quality cutting, intricate designs or low cutability steels. Highest speeds are for lower quality cutting, straight line work and high cutability steels. Width of kerf will vary widely with pressure of oxygen, speed, orifice size, and height of tip above the plate.

Flame cutting has no appreciable effect on the physical and chemical characteristics of low-carbon steels except in the case of forgings. These should be annealed and cut hot because of inherent internal strains. Steels over 0.30 per cent carbon tend to some migration of carbon to the cut surface but preheating and annealing obviate this tendency. Preheating to 200 to 800 F is also essential for maximum ease in the cutting of alloy steels and for heavy sections of large mass except where analysis makes such treatment inadvisable owing to metal affinity for the oxygen jet at elevated temperatures. With stock over 0.40 per cent carbon, total decarburization should not exceed 0.0006-inch per side to obviate undesirable excess slag adherence.

Ordinarily the chromium and nickel content of stainless steels produces oxides which make ordinary cutting slow and undesirable. However, a new method which utilizes a cutting head that injects a flux into the oxygen stream makes possible the cutting of stainless steels of as high as 50 per cent alloy content as readily as carbon steels by the ordinary torch. High quality cuts, equal to those regularly obtained, can be made at production rates in material up to 3 inches thick and rough cuts in sections up to 6 inches.

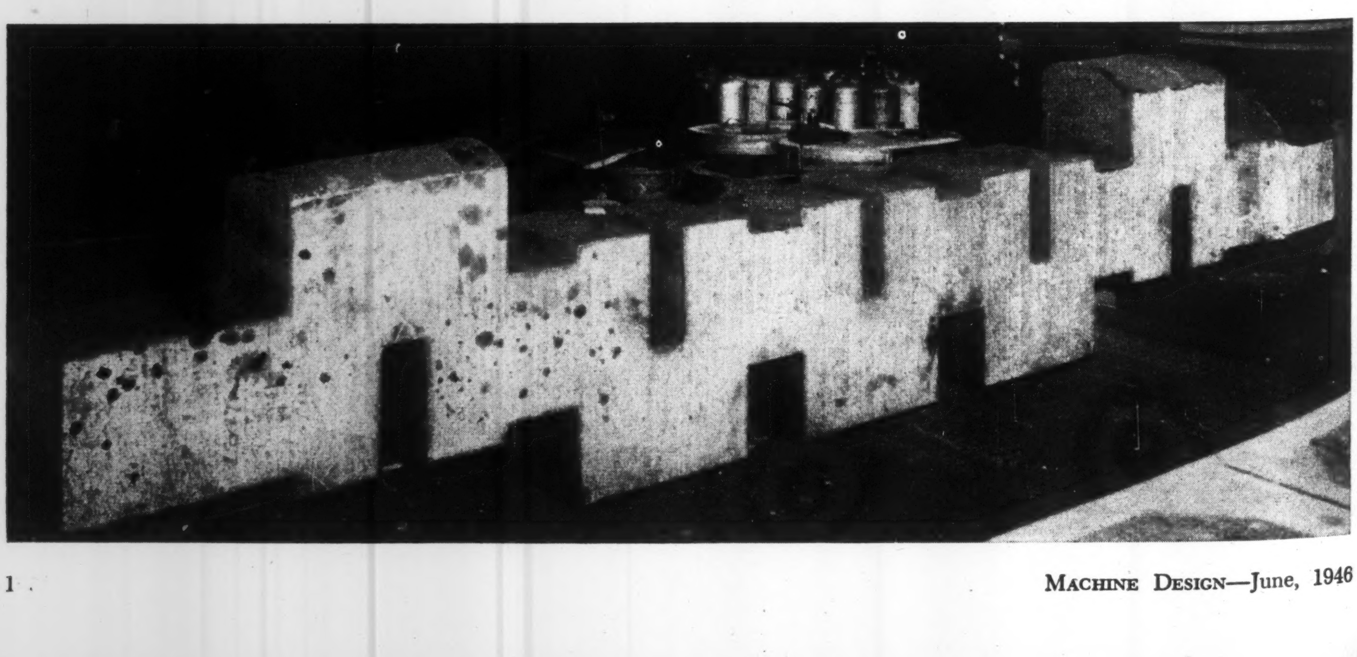
Material to be flame cut should have good surface conditions free from surface pitting, rounded corners, crowning, etc., to assure full utilization of the close tolerances available. Material should be pickled so as to be free from scale and all but moderate rusting avoided.

Tolerances: Final accuracy on a flame-cut part depends primarily upon a number of factors: (1) Accuracy and rigidity of the machine used; (2) accuracy of the oxygen jet; (3) thickness of the material; (4) cutability of the material; and (5) the clamping and distortion factors. Naturally, the most accurate machines only should be required to produce "precision" work. Thus, most portable straight-line machines are recommended only for "structural accuracies", that is where welding type edge preparation is of sufficient accuracy. This usually works out to be within 1/8-inch, plus or minus. Likewise with portable shape-cutting machines, similar and, in certain cases, somewhat better accuracy can be achieved.

Stationary Machines Most Accurate

Stationary shape-cutting machines are far more accurate, and certain ones regularly reproduce contours, guided by a magnetic follower, accurate to 1/64-inch, plus or minus. Flame planers produce straight cuts accurate to plus or minus 1/64-inch or less. As a rule, the oxygen jet will remain within about 0.006-inch of a desired line or contour. Consequently, a combined accuracy of plus or minus 0.021-inch in such cases is about all that can be expected. To this must be added the usual work-distortion and work-movement factors which vary over wide limits from nothing up to a matter of inches, depending upon the cutting technique, material condition, plate size, and procedure control. These can be handled properly and detrimental effects obviated to a great extent as discussed to some extent previously. One manufacturer has found

Fig. 9—Below—General view of a gigantic crankshaft torch cut from a forged slab prior to twisting and machining



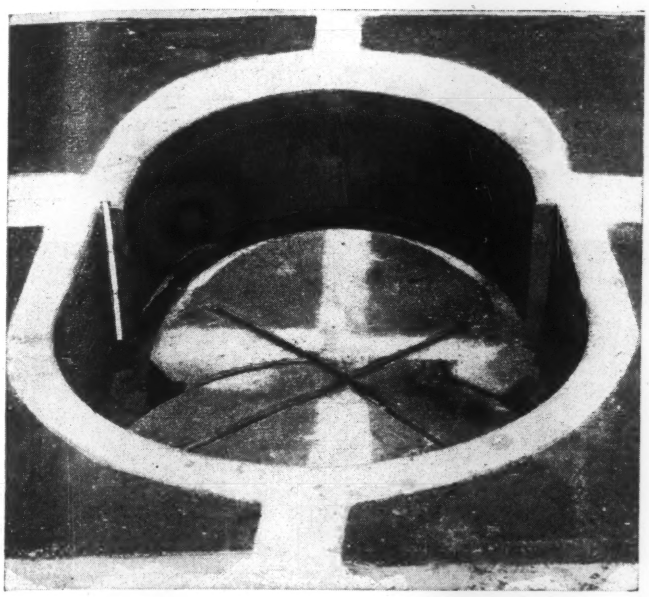


Fig. 10—Above—Contoured hole, cut at 2 to 3 inches per minute in 18-inch armor plate, shows squareness of a 2-inch offset

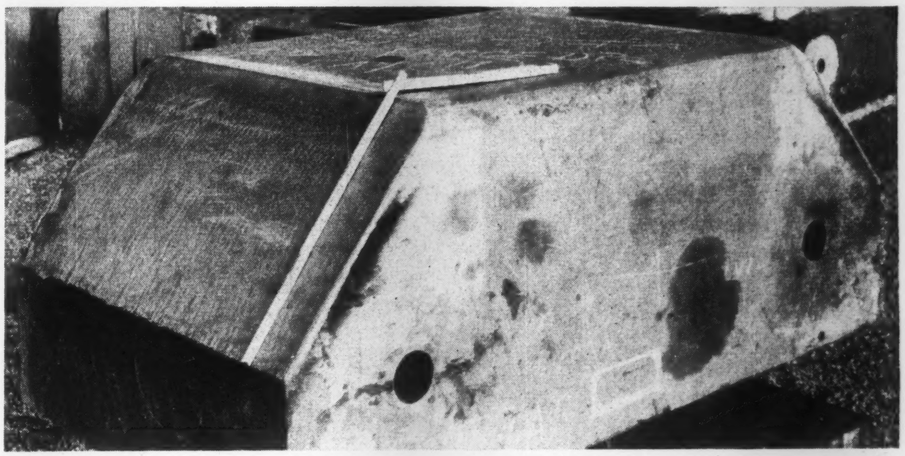


Fig. 11—Right—Bevel cut 17.5 inches long on a heavy die block shows fine surface finish obtainable

that convex distortion on straight-line cuts will not exceed 1/16-inch on a flat plate having a width of 3 feet or greater irrespective of length. Where the cut length is 10 feet or less, or the thickness under 1/2-inch, the distortion seldom exceeds 1/32-inch. Materials 1/2-inch and over in thickness are seldom warped or buckled unless unduly long or narrow. Cases on record show that with an electronic tracing device and the usual sturdy pantograph machine, template or drawing contours can be reproduced accurately to within plus or minus 0.002-inch. Considering the jet inaccuracies, such machines can be expected to produce shapes in quantity with contours identical within plus or minus 0.008-inch, barring other factors. In flame planer work, the total inaccuracy can be consistently held within 0.030-inch on a side with 1/2-inch plate up to 30 feet in length.

Weight Tolerance May Be Used

Close tolerances, of course, should only be required where absolutely necessary. Many applications do not require exacting dimensional tolerances and in such cases the usual procedure is to specify work on a weight basis of 5 per cent tolerance or over.

In addition to variations of the oxygen jet from a line or contour are the variations in the vertical accuracy of the kerf produced by the jet. Ordinarily, cut surfaces true to within plus or minus 0.003-inch as to cross-sectional squareness for steel under 0.35 per cent carbon around 1 inch in thickness can be had. This becomes 0.010-inch for 2-inch material, 0.018-inch for 4-inch material, 0.031-inch for 6-inch material, 0.062-inch for 12-inch material, and 0.125-inch for 24-inch material. These tolerances should not be specified for ordinary work but in cases of necessity can be had and oftentimes bettered, *Fig.* 10, with proper handling.

Extreme accuracy in reproduction of shapes requires knowledge of kerf widths produced. Any template used must be made with the necessary allowances. For "precision" work the rule-of-thumb allowance of twice the oxygen orifice is not sufficiently accurate. Actual kerf widths are determined by the cutting conditions involved and vary appreciably with pressure, speed, tip size, and space from the tip to the plate. Consequently they must be obtained by actual observation for each job and the conditions under which this observation is made adhered to rigidly to avoid variations. An idea as to cutting tip size for various plate thickness, approximate kerf widths, etc. is given in Table I.

Surface smoothness depends primarily upon the rigidity of the torch-holding mechanism and secondarily upon atmospheric conditions. Air movement at the cut may affect the smoothness somewhat. However, wherever the cut surface may constitute a bearing surface, gear tooth, etc., finish machining, broaching or milling is usually necessary. In such cases allowance should be made on the template and with careful cutting it is necessary to remove only 0.002 to 0.003-inch of stock in finishing to final dimensions. The bevel cut shown in *Fig.* 11 illustrates the excellent surface quality which can be obtained on even extremely heavy cuts.

Collaboration of the following organizations in the preparation of this article is acknowledged with appreciation:

Air Reduction Sales Co. (Figs. 1 and 5)	New York
Bethlehem Steel Co. (Figs. 9, 10 and 11)	Bethlehem, Pa.
Indiana Oxygen Co. (Fig. 4)	Indianapolis
Linde Air Products Co. (Figs. 2, 3, 6 and 8)	New York
Lukenweld, Inc. (Fig. 7)	Coatesville, Pa.
Olson Steamship & Navigation Corp. (Figs. 1 and 5)	Pasadena, Calif.
Victor Equipment Co.	San Francisco

Torsion Bar Provides Couplin

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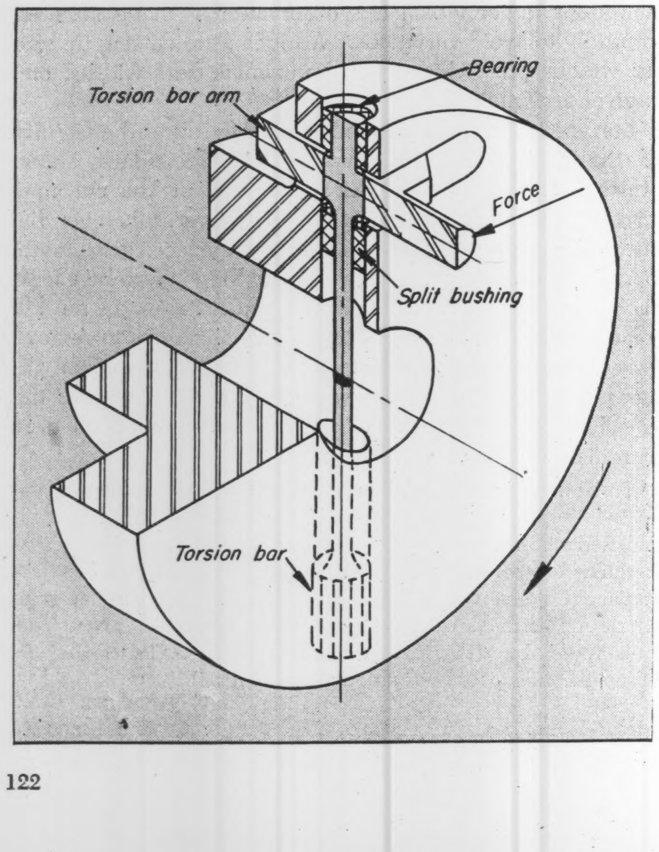
Baltimore

FUNCTIONS of a flexible coupling, in general, are to allow for the linear and angular misalignment which may exist between the shafts of the driving and driven members of a mechanical system[1]. Certain types of flexible couplings also will absorb or dampen vibrations, or will cushion the effects of shock which would otherwise be transmitted.

In the present article the design of a new type of torsionally flexible coupling is discussed and results of a number of tests made on a full-scale working model are presented. The new coupling, views of which are shown in *Figs.* 1, 2 and 3, was designed by Mr. Max Essl of the William H. Harman Corp., Philadelphia.

As shown in *Figs.* 2 and 3, the coupling has the conventional steel flanges keyed to the driving and driven shafts, respectively. It will be noted, however, that the flanges are joined by only a single pin, referred to as the "torsion bar arm". Either flange may be used on the driving side, provided the torsion bar arm is properly adjusted. For the purposes of this discussion it has been assumed that the right-hand flange is on the driving shaft.

Fig. 1—Below—Section showing torsion bar in driving element. Force indicated is reaction from driven element



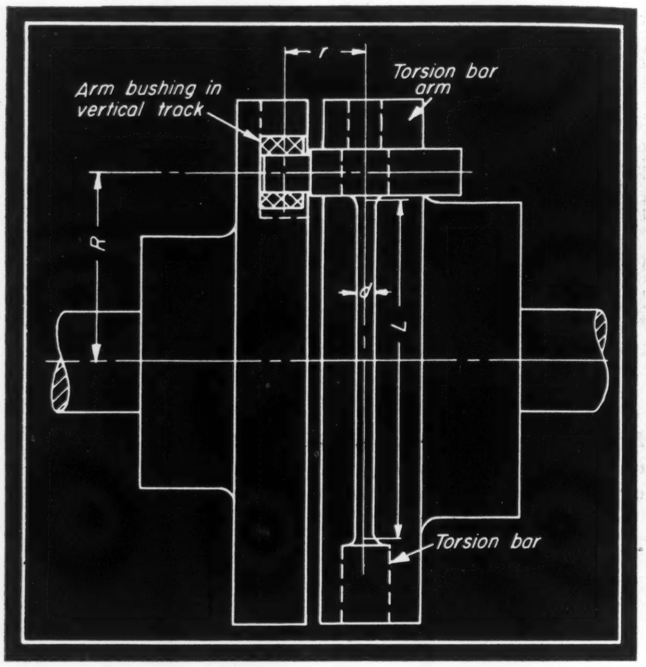


Fig. 2—Elevation in part section shows principal dimensions that control characteristics of the coupling

In the flange on the driven side there is a bronze bearing, referred to as the "arm bushing", which transmits to the driven shaft the force applied by the torsion bar arm. The arm bushing is approximately cylindrical in shape about an axis normal to the axis of the torsion bar arm. Should the two shafts be installed parallel but out of center, the distance *R, Fig.* 2, between the axis of the torsion bar arm and the axis of the driven shaft, would vary during each revolution. To accommodate such parallel misalignment, the bronze arm bushing is set in a guide, so that it is free to move radially in the driven flange. As will be evident from inspection, the arm bushing may move about an axis normal to the axis of the driven shaft.

The torsion bar lies in the driving flange, as shown on *Figs.* 1 and 2, with its axis at right angles to the axis of the driving shaft. Lower end of the torsion bar is splined and is held firmly in the flange. Near the upper end the diameter is increased so that a spline connection can be made with the torsion bar arm, *Fig.* 3. The torsion bar terminates in a journal which fits into a bronze bearing in the flange. To prevent bending deflection of the torsion bar, another bronze split bushing is provided on a journal immediately below the torsion bar arm.

Operation of the coupling is best understood if reference is first made to *Fig.* 3. When the unit is installed, the torsion bar arm is set so that its axis departs from that of the main shaft by an angle α. Assuming that the torsion

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[1] "Specifying Intermediate Components for Machine Drives"—R. K. Lotz, MACHINE DESIGN, Oct., 1944.

inlexibility

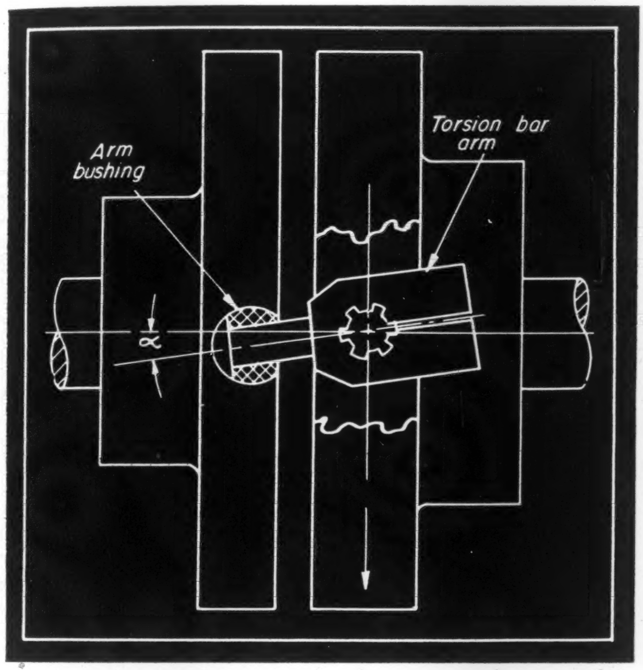


Fig. 3—Coupling is designed so that angle α becomes zero when the rated torque is being transmitted

bar lies in the driving flange, the flange will start to move in the direction indicated by the arrow. With a steady load on the coupling, the torsion bar arm is pulled over until, at rated torque, the angle α becomes zero. This action is accompanied by a slight displacement of the torsion bar arm in the arm bushing. Any variation of torque is reflected in the position of the torsion bar arm, which in turn imposes a proportionate moment on the torsion bar.

Within the elastic limit, the torsion bar acts as a spring. A linear relationship exists between the torque and the deflection, so that it is possible to determine the power transmitted if the speed of the shaft and the deflection of the torsion bar are known.

Analysis of the coupling is as follows. The relation between brake horsepower *bhp*, shaft torque T, (in.-lb) and shaft speed N (rpm) is given by the equation

$$bhp = \frac{2\pi NT}{12 \times 33000} \qquad (1)$$

Twisting moment M (in.-lb) on the torsion bar is given by

$$M = \frac{Tr}{R} \qquad (2)$$

where r is the effective radius of the torsion bar arm, inches, as shown in *Fig. 2*. In terms of the dimensions and characteristics of the torsion bar the twisting moment is

$$M = \frac{I_p S_s}{d/2} = \frac{\alpha G I_p}{L} \qquad (3)$$

where d = diameter of the torsion bar, inches; L = effective length of the torsion bar, inches, (*Fig. 2*); I_p = polar moment of inertia of the torsion bar = $\pi d^4/32$ inches^4; G = shear modulus, psi; S_s = shear stress on the extreme fiber of the torsion bar, psi; and α is the angular deflection, radians, (*Fig. 3*). Combining Equations 1, 2 and 3, the horsepower transmitted may be expressed as

$$bhp = \frac{\alpha G I_p R N}{63{,}024\ Lr} \qquad (4)$$

At a given speed, therefore, the transmitted power is directly proportional to the angle α.

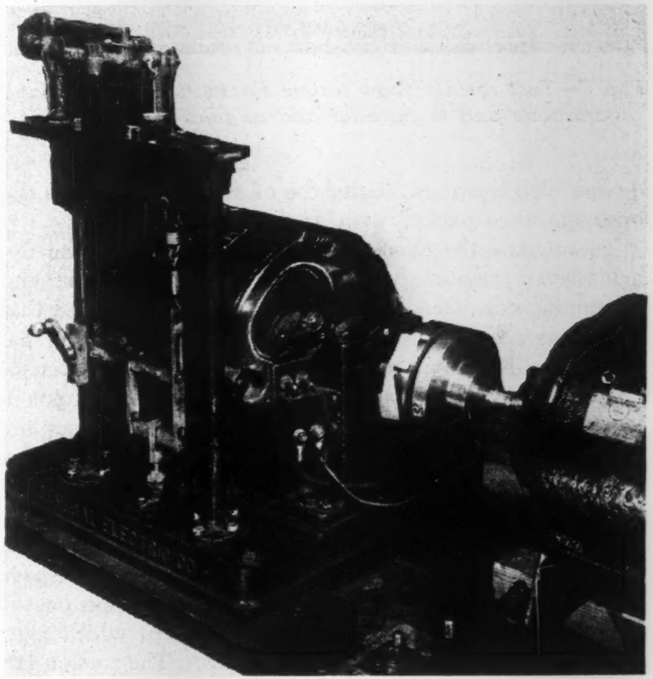
Within the elastic limit, strain energy, U, stored in the twisted torsion bar is equal to [2]

$$U = \frac{\alpha^2 G I_p}{2L} \qquad (5)$$

Equation 5 is both interesting and important, as it indicates the limits within which this torsionally flexible coupling may be considered a vibration damper. For a given length of shaft, the capacity of the torsion bar for the storage of strain energy is fixed by the allowable angle of deflection and the polar moment of inertia of the bar. This last factor depends upon the fourth power of the diameter. Clearly, then, with bars small enough to satisfy the normal torque requirements and limited as to total permissible angles of deflection, only a small amount of strain energy may be stored.

The coupling built for the laboratory tests has mild

Fig. 4—Below—Test setup showing aluminum pointer and scale attached to driving and driven elements



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[2] *Strength of Materials*—S. Timoshenko, Vol. 1, Page 303.

steel flanges 8⅜ inches in diameter. The torsion bar is NE 9442 steel, heat treated to 375-440 brinell hardness number, the diameter being ¼-inch and the effective length 5½ inches. The distance *R*, between the driven shaft and the axis of the torsion bar arm, is 3 inches and the radius of the torsion bar arm is 1 9/16 inches. This coupling is designed to transmit 5 bhp at 1000 rpm.

In order to measure the actual deflection of the torsion bar arm, an aluminum protractor was attached to the edge of the flange, and an aluminum pointer fastened to the torsion bar arm. The pointer and protractor are shown in place in *Fig.* 4. Also shown in *Fig.* 4 is the principal test equipment used in the laboratory. The coupling was installed between a d-c motor and a cradle type electric dynamometer, with the torsion bar in the driven flange.

During the run the protractor was observed with a General Radio "Strobotac", enabling the twist in the torsion bar to be seen. Under several steady load tests the coupling ran smoothly and no trouble was experienced with the oil-filled bearings. Later the oil was removed, and the bushings were packed with grease.

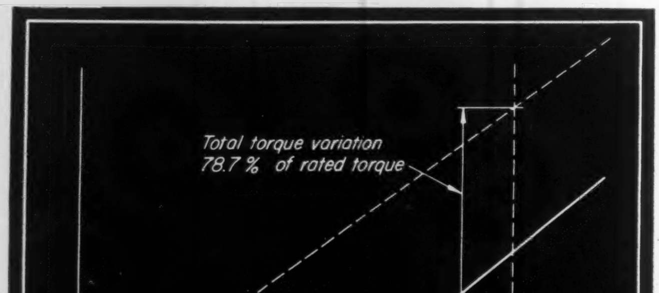

Fig. 5—Test results show torque fluctuations due to load variations and to parallel and angular misalignment

To simulate the effects of variable driven load, the dynamometer armature was supplied with both direct and alternating current. The frequency was so controlled that there was a complete a-c cycle for each revolution of the coupling. The resulting torque characteristic was that of an alternating or pulsating torque superimposed upon a larger steady torque. Under these conditions the coupling functioned smoothly. The Strobotac revealed that the torsion bar was flexing with each torque variation imposed by the dynamometer.

To observe the effect of angular misalignment the motor was moved so that, in the plane of the shafts, the space between the flanges was greater on one side than on the other. The distance chosen was 1/16-inch, which gave an angular misalignment of 26 minutes. The torsion bar showed wider fluctuations under this condition, but the coupling performance was satisfactory.

While still maintaining the angular misalignment, parallel misalignment was introduced by sliding shims under the legs of the motor. Satisfactory runs were completed with parallel misalignment of 1/32-inch combined with angular misalignment of 26 minutes.

In order to show the effect of torque variation on the driving side, a new test unit was prepared, using a large d-c motor and a General Electric inductor dynamometer. Again the test work was completed at 1000 rpm with the linear and angular misalignments referred to above. The coupling was reassembled on the new test unit with only the grease which clung to the parts. In spite of this, there was no difficulty with the moving parts. The detailed technique of the tests was as follows:

1. By the use of Equation 4 the theoretical deflections were computed for chosen values of bhp at 1000 rpm under steady torque conditions
2. A calibration run was made under steady torque at 1000 rpm, and actual pointer positions were observed with the Strobotac
3. Running at 1000 rpm with fluctuating torque, the actual pointer positions were measured at a number of points around the circumference of the flange. From these observations the maximum and minimum deflections were determined, and the maximum and minimum values of torque calculated.

In *Fig.* 5 is shown a plot of torque against bhp under varying torque on the driving side together with parallel misalignment of 1/32-inch and angular misalignment of 26 minutes. The line marked "rated torque" represents the average torque required to produce the bhp registered on the dynamometer. The broken lines indicate maximum and minimum deflections of the torsion bar, referred back to equivalent shaft torque by the methods already outlined. The vertical dotted line represents the design conditions of 5 bhp at 1000 rpm. It is noteworthy that at this loading, the torque variation amounted to 78.7 per cent of the average torque.

Following are the principal conclusions drawn from the tests:

1. The coupling operated satisfactorily even with combined linear and angular misalignment in excess of that to be expected in practice
2. Either grease or oil lubrication was satisfactory
3. Test experience indicated the desirability of using thrust bearings in both the driving and driven members
4. A field should exist for this type of coupling where the torque input is fluctuating and more uniform torque is required for the driven member. This would include applications in the machine tool and internal combustion engine fields
5. The coupling could also be adapted for use as a torque meter, using a pointer and protractor which would read torque directly in pounds-feet. Such a unit would have importance in the field testing of such units as fans and pumps.

The author wishes to thank Dr. C. F. Miller, of the Department of Electrical Engineering, The Johns Hopkins University, for his great assistance with the electrical phases of this investigation.

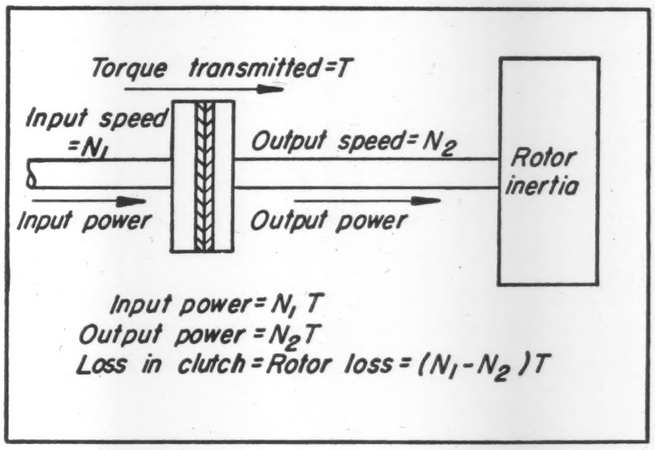

Fig. 1 — Friction clutch analogy for acceleration of an induction motor

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Evaluating Motor Losses through Clutch Analogy

ORDINARILY duty-cycle applications of squirrel cage motors are thought of as requiring frequent starting, stopping, reversing and, on multispeed motors, changing from one speed to another. The only way to apply a motor accurately for this type of service is to calculate the losses in the motor for the various portions of the cycle and see if the losses are within the amount the motor can dissipate within a safe temperature rise. Although such a procedure requires data available only to a motor designer, it is possible to get a picture of the severity of a duty cycle without detailed motor design information.

LARGER MOTORS do not always solve problems involving overheating when frequent plugging stops or reversals are necessary. Inertia losses in a motor, resulting in heating when it is accelerated or decelerated, may be understood and approximated readily through the simple clutch analogy discussed in the accompanying abstract of a paper presented at the recent Westinghouse Machine Tool Electrification Forum

The torque a motor must deliver during acceleration may be considered to consist of two components, one of which supplies the running load and the other the acceleration. On most duty-cycle applications the running torque is small compared to the accelerating torque and may be neglected for estimating purposes. The problem then becomes one of determining losses in an induction motor during the accelerating or stopping of an inertia load.

In an induction motor the stator winding creates a magnetic field rotating at synchronous speed and the rotor operates at some slip speed with respect to the rotating magnetic field. As far as acceleration is concerned, an analogy can be made with a friction clutch in which one plate represents the rotating magnetic field and is driven at constant speed equal to the synchronous speed of the motor, and the other plate represents the rotor and runs at the rotor speed. *Fig.* 1 illustrates this analogy. Rotor inertia will be considered to include the inertia of the rotor plus the inertia of all connected rotating parts. It is assumed that no running load is present.

Input power is equal to the speed of the input shaft multiplied by the torque transmitted by the clutch. Output power is equal to the speed of the output shaft multiplied by the torque transmitted by the clutch. The loss in the clutch is equal to the difference between the input and the output power and is, therefore, equal to the difference between the input and output speeds multiplied by the torque transmitted through the clutch. The difference between input and output speeds is known as the slip speed and, therefore, it can be seen that loss in the clutch is equal to the slip speed multiplied by the torque transmitted. The friction loss in the clutch represents the I^2R loss in the rotor.

Torque transmitted by a friction clutch is approximately constant regardless of the speed whereas torque developed by the rotor of an induction motor varies considerably with speed. Energies involved during speed changes, however, will be the same if the torque is considered to be constant.

Application of the analogy to the acceleration of an inertia load from zero to synchronous speed is shown in *Fig.* 2. At zero speed of the output shaft, the output power is zero and, therefore, the entire input power is lost in the clutch. At synchronous speed the output power equals the input power and there is no loss in the clutch. At intermediate values of speed the loss in the clutch is proportional to the difference in speed of the input and output shafts. Power lost in the clutch is, therefore, given by the straight line drawn from *A* to *C* and the total energy lost in the clutch during acceleration is given by the area *ACO*. The total energy stored by the rotor is given by the area *ABC*. It can be seen that this is equal to the total energy lost in the clutch which gives the well-known result that the loss in the rotor of an induction motor during acceleration equals its stored energy.

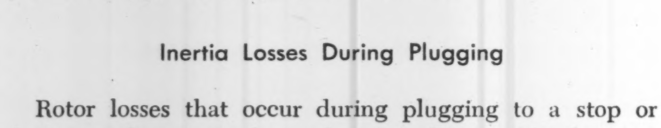
There are, of course, losses in the stator of an induction motor as well as in the rotor. The stator losses may be separated into two components consisting of the I^2R losses and the excitation losses. The excitation losses will be approximately constant regardless of speed while the I^2R loss will be proportional to the rotor loss. The ratio of the stator I^2R loss to the rotor loss is equal to the ratio of the stator resistance to the rotor resistance.

It has been shown that the rotor loss during acceleration must be equal to the stored energy of the rotor. In order to reduce the losses during acceleration, therefore, the only thing that can be done is to try to reduce the stator losses. This can be done by reducing the ratio of the stator resistance to the rotor resistance. It is not possible to reduce the stator resistance appreciably and, therefore, the only way to change the ratio is to increase the rotor resistance. This results in a high slip motor which is the preferred type for frequent acceleration.

Inertia Losses During Plugging

Rotor losses that occur during plugging to a stop or during a plug reversal are shown in *Fig.* 3. At the start of plugging the rotor is running at minus synchronous speed and the difference in speed between the two clutch plates is equal to twice synchronous speed. Since the torque transmitted is constant and the loss is equal to the product of the torque times the difference in speed, the rotor loss is equal to twice that obtained at zero speed of the rotor. If the rotor loss is plotted against the speed the straight line *ACD* is obtained. The total energy loss in the rotor is given by the area *ADF*. Energy stored in the rotor when running at synchronous speed is equal to area *ABC*. It can be seen from the figure that the area *ADF* is four times as large as the area *ABC* which means that the loss in the rotor during a plug reversal is equal to four times the stored energy of the rotor. Loss during a plug stop is given by the area *OCDF*. This means that the loss in the rotor for a plug stop is equal to three times the stored energy of the rotor.

Fig. 2—Rotor output and loss during acceleration of an induction motor


Rotor input
Syn.
A
B
Speed
Rotor output
Rotor loss
O
Power
C
Rotor stored energy = Inertia x (rpm)^2
= Area ABC
Rotor loss during acceleration = Area ACO
= Rotor stored energy

It will be noted that the area *CDE* is equal to *ABC* which represents the stored energy of the rotor at synchronous speed. Area *CDE* represents the energy stored in the rotor at the start of plugging. The loss represented by this area therefore, comes from the stored energy of the rotor and the loss represented by the area *ACEF* comes from the power lines.

Supposing a two-speed motor is plugged from high speed in one direction to half speed in the opposite direction, *Fig.* 4 shows the loss that occurs. If the rotor losses are plotted the straight line *ACD* is obtained. Area *ADF* represents the rotor loss when plugging from full speed to half speed in the other direction. The stored energy of the rotor at the beginning of plugging is represented by the area *CDE*. By comparing areas, it can be seen that the loss in the rotor is equal to 2¼ times the stored energy of the rotor. For a plug stop the rotor loss is equal to two times the stored energy of the rotor. If a two-speed motor is plugged to a stop, a rotor loss will result that is equal to two times the rotor stored energy if plugged on the low-speed winding as compared to three times the rotor stored energy if plugged on the high-speed winding.

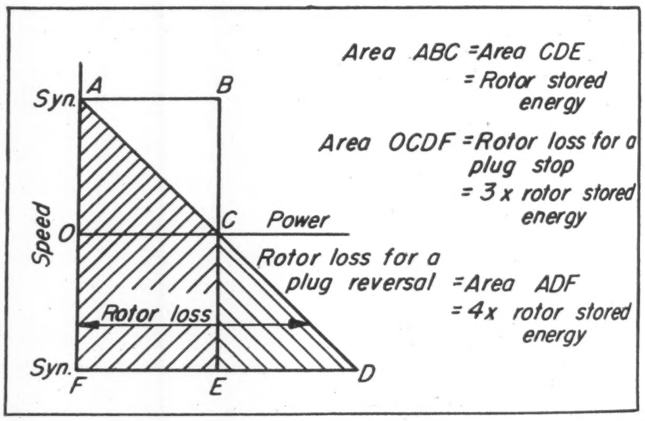

Area ABC = Area CDE
= Rotor stored energy
Area OCDF = Rotor loss for a plug stop
= 3 x rotor stored energy
Syn.
A
B
Speed
O
C
Power
Rotor loss for a plug reversal = Area ADF
= 4x rotor stored energy
Rotor loss
Syn.
F
E
D

Fig. 3—Rotor losses during plugging to a stop and during a plug reversal of an induction motor

Loss involved in a two-speed motor when going from high to low speed by regenerative braking is illustrated in *Fig.* 5. At the first instant after changing from the high to the low-speed winding the rotor is revolving at twice synchronous speed. In the clutch analogy this is equivalent to the rotor plate revolving twice as fast as the stator plate. The speed of the stator plate equals the slip speed of the rotor plate and the rotor loss at high speed is equal to the stator power. In *Fig.* 5 the rotor loss at high speed is represented by *AB* and the stator power by *BC*. Since the rotor plate is tending to drive the stator plate the torque transmitted by the clutch is reversed from the normal motoring direction and the stator power is regenerated. The power regenerated will remain constant with respect to speed while the rotor loss will be equal to the horizontal distance between the lines *BD* and *AD*. Stored energy of the rotor is represented by the area *ACO*. The loss in the rotor when regenerated from high to low speed will be represented by the area *ABD* and is equal to ¼ the stored energy of the rotor. Power regenerated is represented by the area *BOED*. This is equal to ½ the stored energy of the rotor. The area *DEO* represents the remaining energy stored in the rotor at half speed and is equal to ¼ of the energy stored in the rotor at top speed.

Losses involved in a four-speed motor during regenerative braking is shown in *Fig.* 6. Assuming that the rotor is at high speed and the stator is suddenly connected for ¾ speed, the rotor slip speed is then 1/3 of the stator speed and the rotor loss is 1/3 of the stator power. In *Fig.* 6, *AB* represents the rotor loss and *BC* the stator power regenerated. Area *ABD* represents the rotor loss and area *BCFD* the regenerated energy while braking regeneratively from high to ¾ speed. Area *ACO* represents the stored energy of the rotor at high speed. Comparing areas, the rotor loss is found to be 1/16 and the regenerated energy 3/8 of the stored energy of the rotor at top speed. If the motor is braked to half speed in two steps the rotor loss will be 1/8 of the rotor stored energy. This is 1/2 of the rotor loss that occurs when regenerating to half speed in one step as shown in *Fig.* 5.

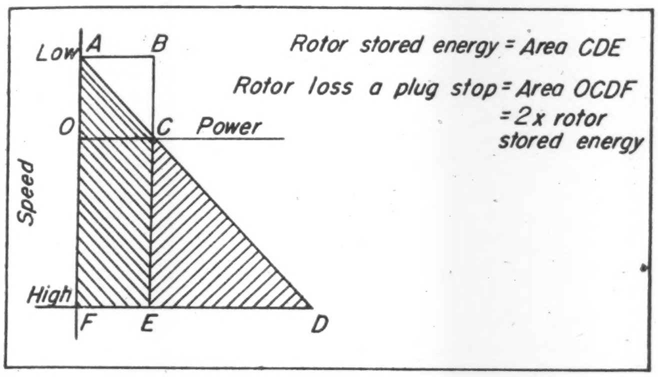

Low
A
B
Rotor stored energy = Area CDE
Rotor loss a plug stop = Area OCDF
= 2x rotor stored energy
O
C
Power
Speed
High
F
E
D

Fig. 4—Plugging of an induction motor from high speed in one direction to half speed in the opposite

In all of the above examples the losses in the rotor depend directly on the total inertia of the parts direct-connected to the motor shaft. In designing a machine for duty-cycle operation, the rotating parts such as sheaves, gears, chucks, etc., should be designed to have the minimum inertia when direct-connected to the motor shaft. The inertia of these parts often can be reduced considerably by suitable design. Importance of this point cannot be overemphasized as the number of cycles per hour of which the motor is capable depends largely on the stored energy of the rotating parts.

Ventilation of the motor is an important factor on duty-cycle applications. The losses an open motor can dissipate drop off considerably with speed due to a decrease in the amount of air forced through the motor. A motor that starts and stops or reverses frequently, may have a low average speed which, with the high losses, will cause the motor to overheat. Assuming that everything possible has been done to decrease the losses during the cycle, the only alternative is to increase the ability of the motor to dissipate its losses. Forced ventilation is the obvious answer. It is very likely that forced ventilation will be used on many of the more difficult applications.

A word of caution is probably in order at this point. If a motor overheats on an ordinary application the natural reaction is to put in a larger motor. This does not necessarily work on duty-cycle applications. For example, considering a reversing application with negligible connected inertia and running load, increasing the motor rating will result in a decrease in the number of reversals obtained. The reason for this is that as the motor frame size is increased, the stored energy in the rotor increases faster than the dissipating ability of the motor.

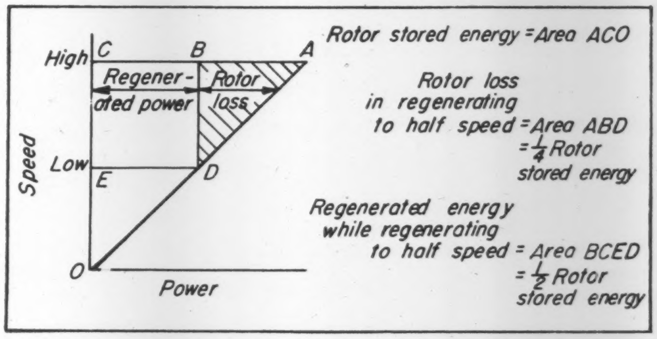
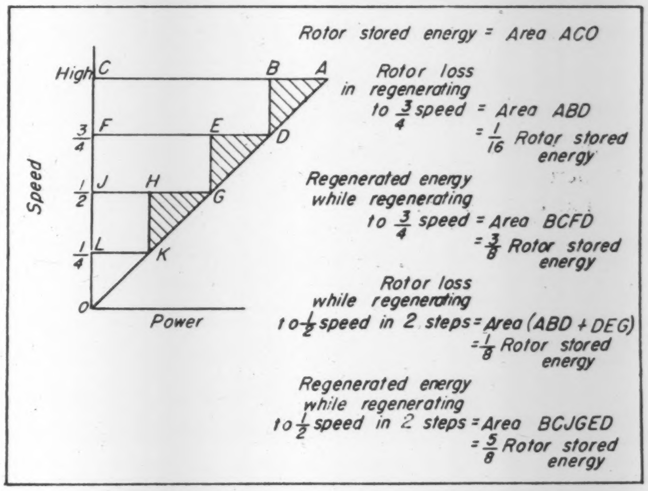

High
C
B
A
Regener-ated power
Rotor loss
Low
E
D
Speed
O
Power
Rotor stored energy = Area ACO
Rotor loss in regenerating to half speed = Area ABD = 1/4 Rotor stored energy
Regenerated energy while regenerating to half speed = Area BCED = 1/2 Rotor stored energy

Fig. 5 Above—Regenerative braking from high speed to low speed with a two-speed induction motor

Fig. 6 Below—Regenerative braking with a four-speed induction motor, showing energy saved


Rotor stored energy = Area ACO
High
C
B
A
3/4
F
E
D
1/2
J
H
G
1/4
L
K
O
Speed
Power
Rotor loss in regenerating to 3/4 speed = Area ABD = 1/16 Rotor stored energy
Regenerated energy while regenerating to 3/4 speed = Area BCFD = 3/8 Rotor stored energy
Rotor loss while regenerating to 1/2 speed in 2 steps = Area (ABD + DEG) = 1/8 Rotor stored energy
Regenerated energy while regenerating to 1/2 speed in 2 steps = Area BCJGED = 5/8 Rotor stored energy

Another warning: Do not use Class II motors on duty-cycle applications. These motors have a double-cage rotor construction and are intended for applications requiring infrequent starting, high starting torque, and good running characteristics. The double-cage construction consists of large, low-resistance bars imbedded deep in the iron and small, high-resistance bars near the surface of the rotor iron. At low speeds the small bars carry most of the rotor current, thus giving a high resistance characteristic for starting. At high speeds the large bars carry most of the current, thus giving a low rotor resistance for good running characteristics. Most of the rotor starting losses are concentrated in the small bars and, if the motor is started or reversed frequently, the rotor will fail mechanically due to unequal expansion of the large and small bars.

A few of the fundamentals on the use of squirrel cage motors on duty-cycle applications have been reviewed. The important point is to recognize the types of operation that cause high motor losses, such as accelerating, plugging, and regenerating, and to check the application with a motor designer if these operations occur frequently.

Bearing Failures—Causes and Cures

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Part II

PARTIAL lubrication defines a wide range of types of lubrication, from adsorbed films of molecular dimensions to much thicker films of nearly the thickness obtained with hydrodynamic lubrication. Conditions are much more severe with thin films, and the speeds and loads must, therefore, be kept lower. The design should be such as to provide the greatest possible thickness of the lubricant film.

Types of failures encountered are galling, wear and scoring. Scoring—gouging of the surfaces—may be localized wear due to grit, or galling, in which particles become attached to one of the surfaces and dig into the other. Therefore, it is necessary to discuss only galling and wear. Both continuous and reciprocating movement will be considered, the first part of the discussion being devoted to continuous movement.

Galling and Surface Temperature

GALLING AND ITS PREVENTION: Galling is welding between the surfaces. Normally one thinks of heat as a necessary factor in welding, but it is not certain whether high temperatures are necessary for galling. The evidence is that they are not.

Nevertheless, heat is certainly a factor under some conditions, and surprisingly high surface temperatures can be reached. Bowden (20)*, investigating surface temperatures at various rubbing speeds, found that high speeds and quite high loads were required to obtain high surface temperatures in sliding, but these could be as high as the melting point of copper-base alloys. He later attempted to extend this work and prove the existence of high surface temperatures in all sliding. In this case, he investigated the temperature flashes in a piece moving by jerks over a surface, and measured only small flashes of temperature during the periods of slipping (21). He speculated on the possibility that these implied much higher surface temperatures, and concluded that they might. Muskat (22), however, in a later investigation of the same effect, concluded otherwise. Considering this work, it appears that high surface speeds are necessary for high surface temperatures and, since galling can occur even at low speeds, high surface temperatures are not necessarily a factor.

Galling is influenced predominantly by the extent of the films on the surfaces. If the two surfaces are chemically clean and freed of all adsorbed film, high friction and galling are almost inevitable, as has been shown by Hardy (23), Langmuir (24), Macauley (25), and by Ernst and Merchant (26). Thus, it is de-

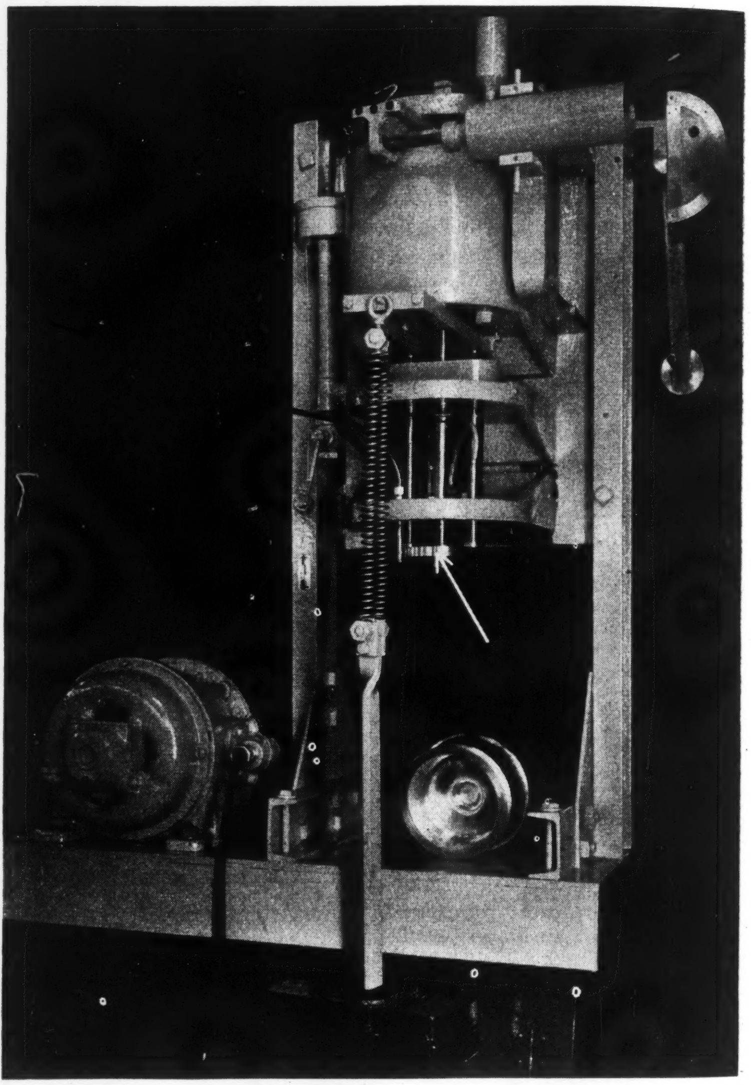

Fig. 4—Machine for testing the performance of bearing materials in the partial lubrication range. Arrow points to specimen in place. Container resting on the table of the machine enables any lubricant, abrasive or atmosphere to be used

sirable to keep surfaces out of contact as far as possible, by a thick oil film if possible, by a thin oil film if conditions do not permit a thick one, or, at the very least, by some surface film which can be depended upon to be present at all times. If this is not accomplished, trouble is inevitable.

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*Numbers in parentheses refer to references listed at end of article.

A film of lubricant is most desirable. This is obtained best by grooving, and the best grooves are those which enable the lubricant to reach all of one of the two surfaces. Their effect under these circumstances is highly beneficial and, under some conditions, provides a close approach to hydrodynamic lubrication, as Fogg (27) has shown for thrust washers. Teetor (28) also discusses some of the beneficial effects of interrupting this kind of a bearing surface.

Design of Grooves

Thrust washers used in some aircraft engine accessories are grooved in this fashion and perform well. The oil grooves, as always, should be carefully blended into the surrounding surface so that viscosity effects will assist chemical effects in providing lubrication (29). Incidentally, it appears that rough surface finishes may also provide better lubrication in some instances (30).

Even in the absence of appreciable hydrodynamic effects, with suitable means for getting the lubricant to one of the surfaces, chemical effects can greatly promote lubrication. Extensive studies of adsorbed lubricant films by Bowden (31), Langmuir (32), Adam (33), and Beeck (34), have shown remarkable effects of films even one molecule thick.

In some cases, as with gears, pressures are too high for any of the kinds of lubrication which have been discussed to be effective. When the gear contact is chiefly rolling, as with spur gears, the problem is rarely serious and, as Gatcombe (35) has shown, hydrodynamic lubrication is then at least partially effective. However, with some types of gears the contact is predominantly sliding and then, even though the bearing surface is effectively interrupted, galling (or scuffing) is encountered. In these cases it is necessary to use a chemically active type of lubricant, which coats the gear surface with an adherent nonmetallic film (36). The mechanism of the action of these lubricants appears to be that, when the surface is heated by friction, it reacts with the lubricant to form such a film.

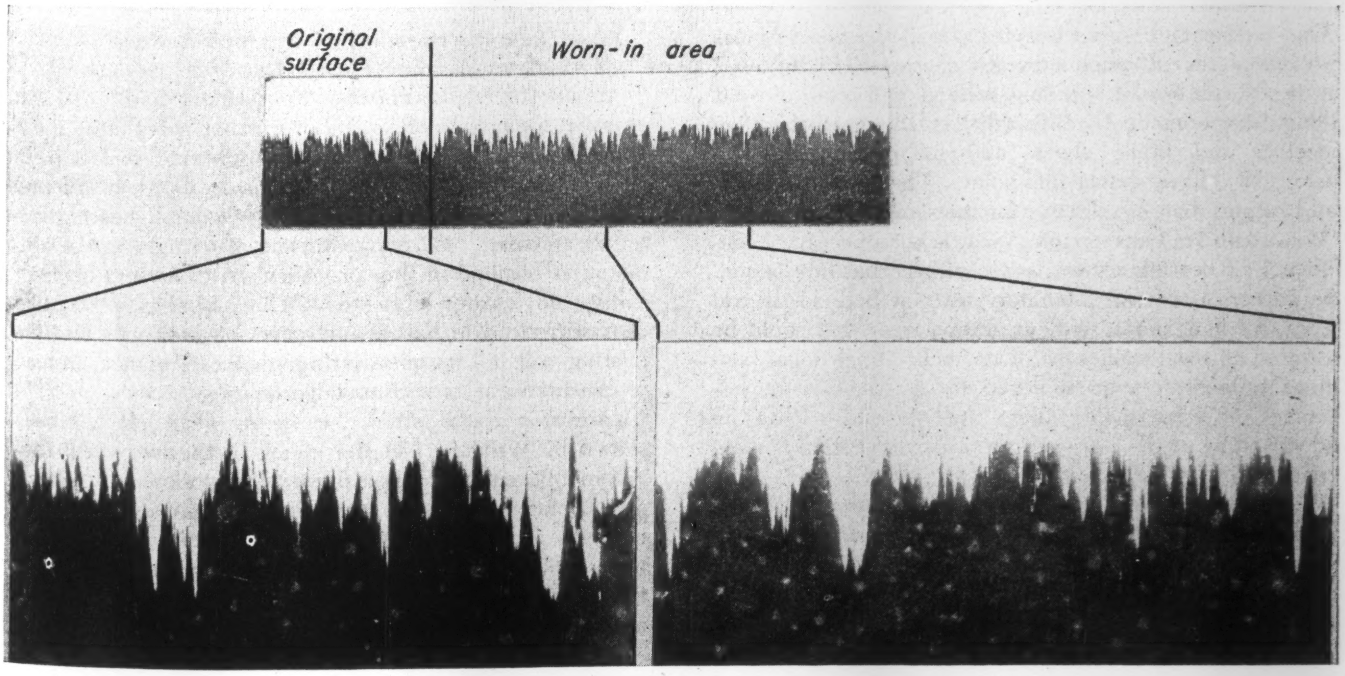
This mechanism shows why E. P. lubricants are not effective in preventing galling when speeds and loads are low, for then surface temperatures high enough to cause chemical reaction are not reached. To prevent galling in such cases, it would be necessary to use some film-forming material that would react at lower temperatures, or one that would act by adsorption (34).

With especially severe conditions, even such lubricants are sometimes not effective in preventing galling, and it is then necessary to use a pair of materials which differ from each other enough so that the galling tendency is reduced. Regardless of the application, the least difficulty with galling is had from a combination of dissimilar materials (37). Similar materials have an affinity for each other that makes welding easy.

Silver, babbitt, bronze, and cast iron all are fairly galling resistant against steel, in about the order named. Since only the surface is important, thin coatings of galling-resistant materials can be used, as tin-plating is used on pistons.

Even better, when circumstances permit their use, is a combination of a metal and a nonmetallic material, for these are still more dissimilar than any pair of metals. Various plastics and nonmetallic compounds have been used. As with metals, thin coatings of nonmetallic materials can be used, such as oxide or phosphate coatings on

Fig. 5—Changes in a shaft produced by wearing-in, as shown by taper sectioning. Magnification of upper micrograph is 100× horizontal and 2500× vertical, which is insufficient to show difference between the original surface and the worn-in area. Two lower photographs, at five times the magnification of the upper, show that wearing in has slightly rounded off the tops of the highest peaks



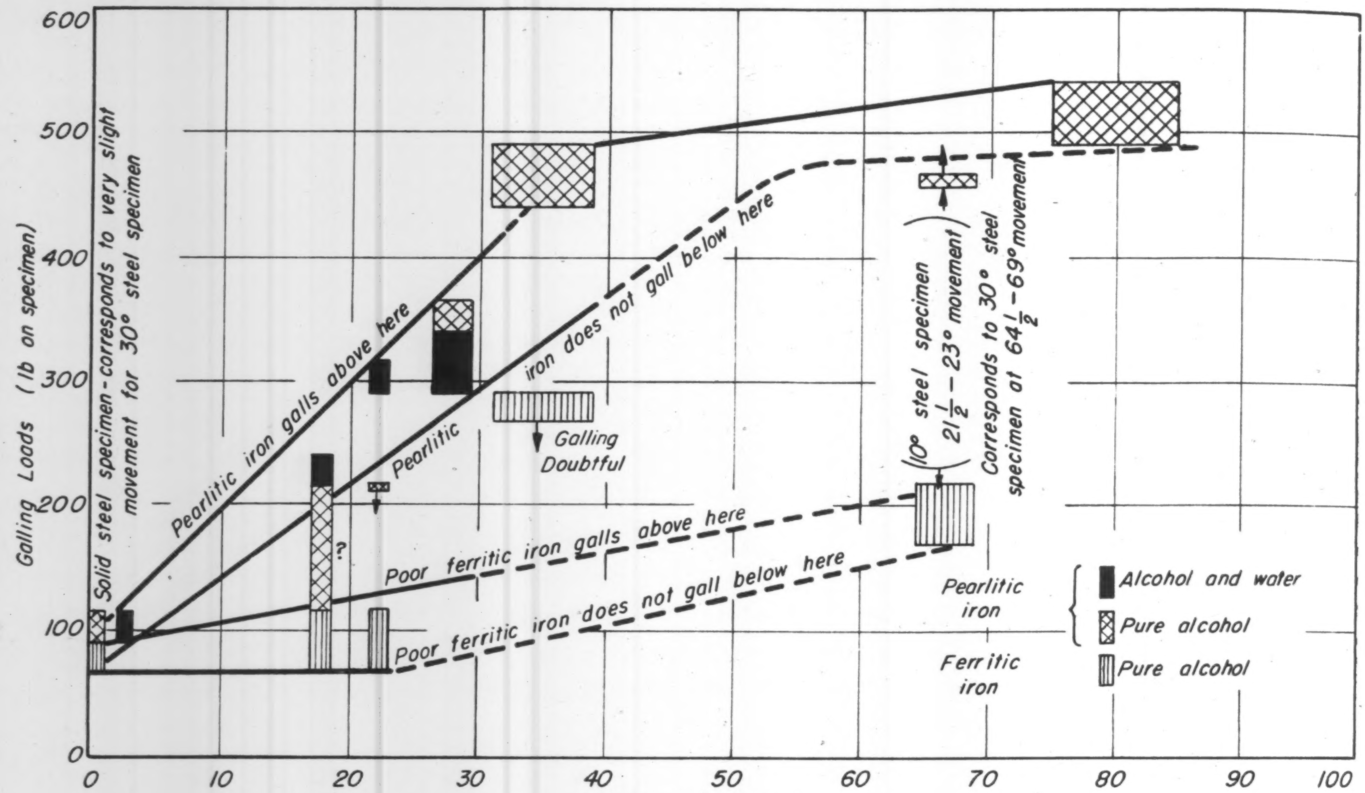


Fig. 6—Relation between the galling load and the angle of reciprocating movement for two kinds of cast iron. Dimensions of specimen are such that 30 degrees of movement uncovers one of the specimens each stroke

piston rings, which are said to give better performance in the critical breaking-in period (38).

Sometimes it is not possible to use a galling-resistant combination, in which case it will be found that harder materials will perform better. It appears that galling is connected with ductility in some way, and that brittle materials resist it best. Thus the very ductile materials, such as stainless steel, are especially bad.

One further fact is that inlaying one of the two surfaces with a material of much different character, so disposed that it will rub on the opposing surface, will often prevent galling. Experiments by Saito (39) on the wear of trolley collectors and brake shoes, and on piston rings by Teetor (28), have shown this point. The lead in copper-lead bearings may be effective for the same reason.

WEAR AND ITS PREVENTION: Wear is not a necessary consequence of bearing service, even with partial lubrication. The authors have run laboratory tests at high loads and speeds, for long times, without getting wear that could be measured in even millionths of an inch. Such ideal conditions cannot often be achieved in service, but the advantages of approaching them are obvious. With an understanding of the causes of wear, its prevention is often apparent.

Wear may be a result of any of several factors (37) (40), galling, abrasion, surface finish, or corrosion. Galling is generally called wear only when it occurs on a microscopic scale, and the torn-out pieces rubbed off afterward (41), so that there is no appearance of galling. The cures are the same as for any case of galling.

One of the most common causes of wear is dirt which, in even tiny amounts, can have a large effect. The best way to avoid it is to keep the dirt out, as ball and roller bearing manufacturers insist. In many cases this is the only solution. In others, where the application permits, at least one of the two surfaces can be made considerably harder than the dirt, and it is then found (42) that wear on both surfaces is small. This is true because the dirt is embedded in the softer surface, so it does not wear it, and the hardness of the dirt is insufficient to wear the other surface.

Relative Hardness of Dirt and Surface

An alternative to making the surfaces hard, to resist abrasive wear, is to make one of them so soft that it is unable to push the dirt against the opposing surface with enough force to abrade it. Thus, in bearings, it is found that there is less damage from dirt when soft bearing materials are used. Carrying this line of thought to the ultimate has resulted in ship propeller shaft bearings made of rubber (43), water lubricated. The rubber is so soft that dirt, stirred up in harbors and rivers, which gets into the bearing will roll harmlessly through the clearance, instead of causing wear as with metallic bearings.

Corrosion too is often a cause of wear. It has been shown by Williams (44) that much of the wear in British automobile engines was a result of corrosion. In a bearing application, many media become corrodents which normally are not, because the rubbing action removes protective films which otherwise would slow down or stop the corrosion. Presumably, both the abrasion and the corrosion resistance of chromium plate are the reasons for its successful application on piston rings and cylinder bores of internal combustion engines (45).

Although wear can be caused also by rough surfaces, the magnitude of wear in this case is generally insufficient to be a problem, except when there is an unusually great disparity of hardness between the two surfaces(37). Wearing-in of a surface soon rounds off the highest peaks of a surface so that wear ceases. Taper sectioning(46) has been used to study the wearing-in process, with the results shown in *Fig.* 5(47).

Difficulties with Reciprocation

Reciprocating Movement: The most difficult type of bearing service is that in which the motion is reciprocating. This covers a wide range of conditions, from those of the piston ring, which moves back and forth a large distance compared with its size, to the press fit which is not supposed to move at all, but often does a tiny bit.

Effect of Motion on Galling: Piston ring action resembles the case of continuous movement, and the discussion of that subject applies here. At the other extreme, when the amount of motion is small with respect to the size of the surfaces, totally different and far more severe conditions are encountered.

This effect is shown well by *Fig.* 6, which illustrates the relation between galling load and the amount of reciprocation. The specimens used for this work are illustrated in *Fig.* 7. The left-hand specimen is cast iron and the raised face, 1-inch inside diameter and 1¼-inch outside diameter, is the one which is tested. The right-hand specimen is fully hardened steel and has three equally spaced lands, ⅞-inch inside diameter and 1⅜-inches outside diameter, each extending 30 degrees circumferentially. The running surfaces are lapped flat before each test.

Pearlitic Iron More Galling Resistant

In testing, the raised surfaces are placed together under load, and the left-hand specimen reciprocated angularly about the center hole. It is apparent that when the angular reciprocation is 30 degrees or more every part of the test surface of the left-hand specimen is exposed to the oil in which the specimens are submerged once each cycle. Two kinds of cast iron—fully pearlitic and ferritic—were tested against fully hardened steel. The first of these is considerably more galling resistant than the second, as the data in *Fig.* 6 show.

Most interesting feature of these data was that the galling load remained nearly constant, as the angle of reciprocation was reduced, until it reached the same value as the specimen dimensions. Then the galling load dropped sharply to very low values at small angles of reciprocation. Incidentally, the specimen area was about one-ninth of a square inch, so that the maximum galling loads represented in these tests are in the neighborhood of 6000 pounds per square inch.

These data were inadequate for small amounts of reciprocation, so another machine was used to extend these studies. This was simply a magnetically operated vibrator, the table of which held one specimen and reciprocated it with an amplitude of about 0.015-inch at 7200 cycles per minute, against another rigidly held specimen, *Fig.* 8. The specimens were one-half inch in diameter, with their ends lapped flat and in contact under light loads. In this test, it was found that any pair of specimens chosen could be galled, even steel and bronze, at a load of only 20 pounds per square inch.

When this galling load is compared with the galling load at large amounts of reciprocation, it is obvious that these conditions are indeed severe. As the amplitude of reciprocation is still further reduced, to the order of microinches of movement, conditions are still more difficult. As Tomlinson(48) has shown, under these conditions any two kinds of material, such as chromium and glass, or steel and wood, will gall.

The appearance of the galled surfaces depends, to a large extent, on the kind and hardness of the surfaces. In general, ductile and soft materials will show more signs of tearing than hard ones, and smaller amounts of motion will show less tearing and more fine debris and oxidation. This phenomenon is called by many names, including fretting corrosion, chafing, brinelling, as well as galling.

Effect of Small Movement on Ball Bearings

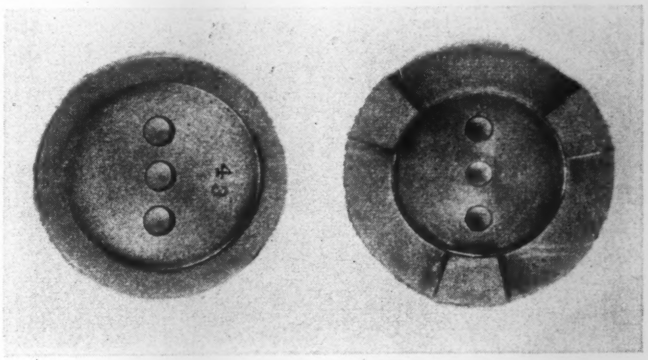
One particular type is the so-called brinelling of ball and roller bearings, when loaded with slight reciprocation for a considerable time. This is a perennial problem to the automobile industry, and Swigert(49) and Almen(50) discuss some of the difficulties of this problem.

The characteristic of reciprocating motion, which causes bearing surfaces to fail so readily, is the scrubbing action with its strong tendency to remove all protective films that are between the surfaces. The surfaces are thus cleaned and brought closer together, and galling is a natural consequence.

In general, under these conditions galling occurs without excessively high temperatures, because the surface speeds are low. Thus, galling usually cannot be prevented by ordinary E. P. lubricants, which require rather high temperature to be effective. However, there are some indications that fluids which are chemically active at ordinary temperatures are an aid in preventing galling, though even they are not effective at the smallest amounts of reciprocation.

Prevention of Galling: When the reciprocating motion is fairly large compared to the size of the specimens, galling is prevented by the same things as when the motion is continuous. As the amplitude of reciprocation is

Fig. 7—Typical specimens used in rotary reciprocation studies. Left-hand specimen is the cast iron to be tested and right-hand specimen is fully hardened steel



reduced to intermediate values, the best solution is to interrupt one surface so that the individual bearing areas are short in the direction of motion, compared to the amplitude of reciprocation.

When the movements are extremely small, no solution can be guaranteed. In an extensive study of galling in aircraft engine parts, Gray and Jenny(51) found that some materials were better than others, especially lead-plated, grit-blasted steel, but that nothing could be altogether depended upon, as long as some slight movements between the surfaces still existed.

Some approaches to this problem have been at least partially effective, but these are specific to the individual problem. Some discussion of the lines of thought may be helpful in arriving at a solution of a new problem.

Preventing Surface Contact

One of the long-standing problems has been galling between the soft steel back of an aircraft master rod bearing and the bore of the rod. This used to be a press-fit joint, but the loads were so high that slight rubbing at the joint occurred. This produced galling, and the galled spots constituted notches which originated fatigue cracks. The best solution to this has been to use a bushing with a few thousandths of an inch clearance in the rod and to key it against rotation. Such a construction has other advantages, and does reduce the galling. It appears quite likely that the effect of a rotating load in promoting hydrodynamic lubrication, as suggested by Underwood(11), plays a part in this success. This approach is seen to be one of preventing contact between the surfaces.

Another approach to the problem is to prevent motion. For example, with an axle which is press-fitted to a wheel, as in railroad practice, it is found that chafing occurs at the press fit, leading to fatigue failure of the axle. Horger(52) found that notching the axle near the wheel, so that the lines of stress were deflected, gave much longer fatigue life. It is a possibility that this approach distributed the shearing force at the joint, and so reduced the magnitude of slippage.

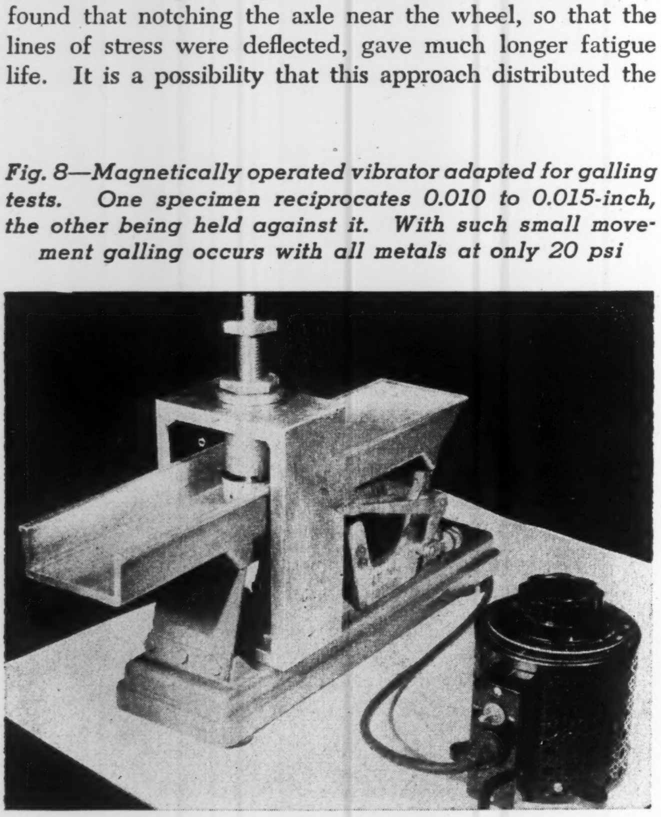


Fig. 8—Magnetically operated vibrator adapted for galling tests. One specimen reciprocates 0.010 to 0.015-inch, the other being held against it. With such small movement galling occurs with all metals at only 20 psi

In many cases the damage to the surfaces is less important than the effect on fatigue strength of the notches produced. In these cases, shot peening may help because, as Almen(53) has shown, the effect of notches is reduced by putting the surface in compression. On the whole, mating surfaces of this kind are among the most troublesome, and any device which the engineer can use to avoid them is worth while.

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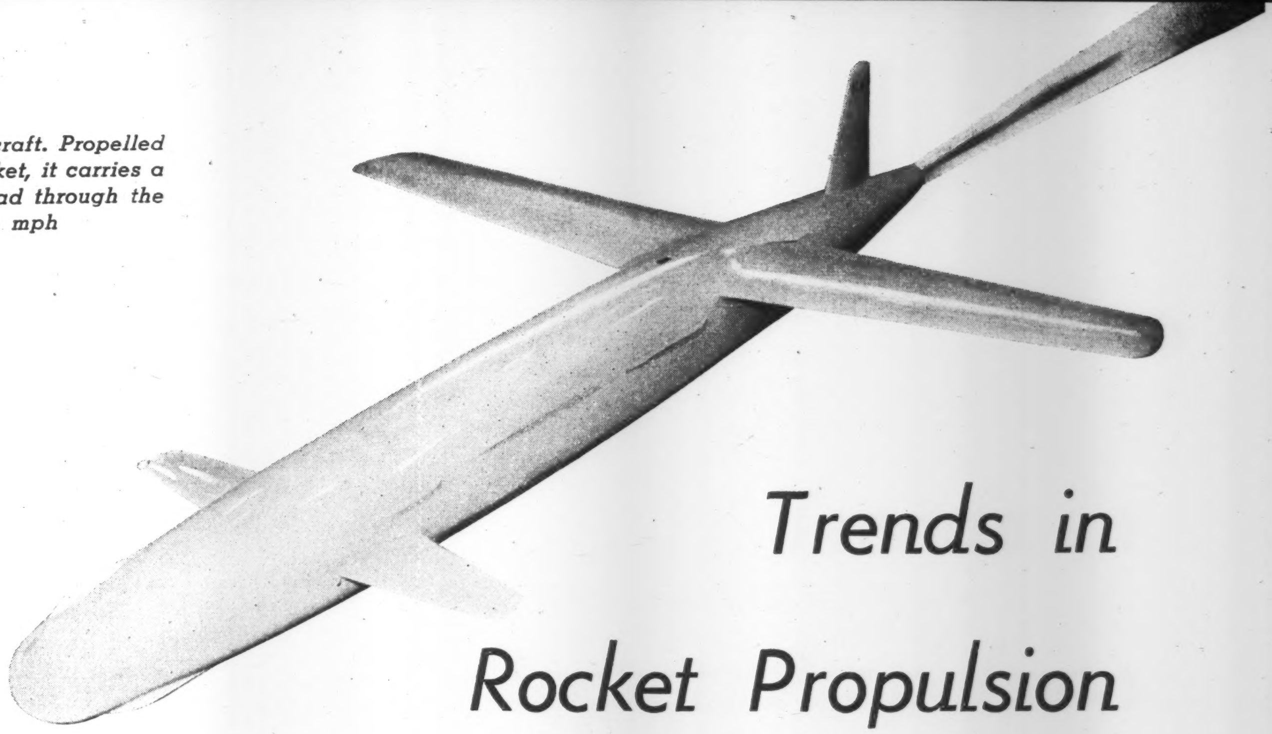


Gorgon pilotless aircraft. Propelled by a liquid-fuel rocket, it carries a 100-pound bomb load through the air at 550 mph

Trends in Rocket Propulsion

By Harry Raech Jr.

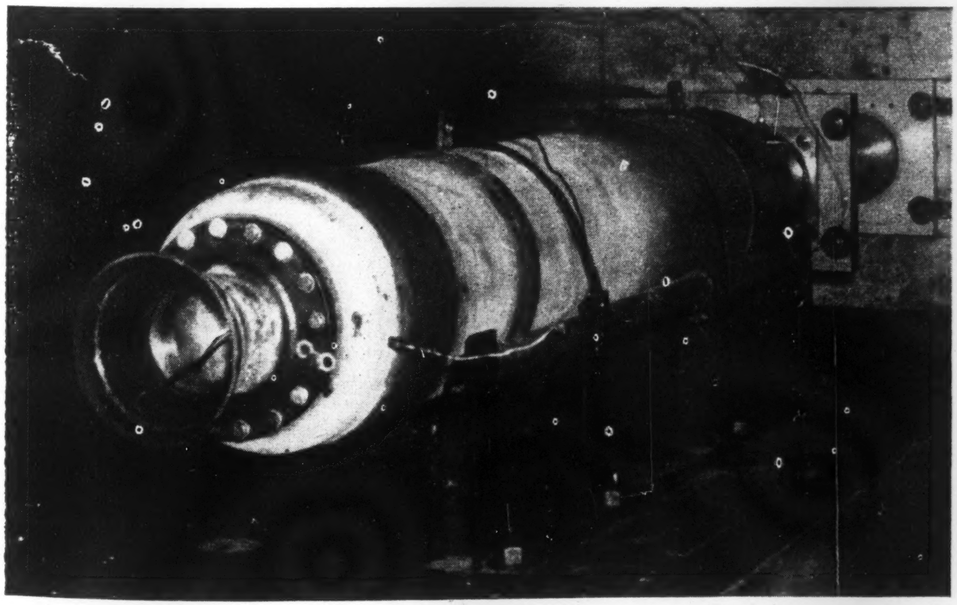
Assistant Editor, Machine Design

OF THE MEANS of propulsion by generated power, rocket reaction is by far the oldest. Yet the preponderance of the development work upon the system has been done within the last ten years. Few stages of our civilization have not known and employed rockets. In India they were used as weapons, in China as fireworks. Rocket combat troops were an integral part of the United States Army during the last century. Still, it was only because of the work of Dr. Robert Goddard two decades ago that the exploitation of the rocket in the recent war was made possible.

Until recently, the rocket principle was used as a means of propelling a short range, erratic projectile, as in the case of the shoulder-fired "Bazooka." It was only after several years of work, largely abroad, that sizable, long-range rockets such as the German V-II bomb, and the Me-163 interceptor plane were developed.

It is in these later developments—promising thrusts to be measured in tons rather than in pounds—that there can be seen a promise for commercial applications. Two basic types, solid and liquid fuel, have seen realization. Rockets propelled by the solid type of propellant were used during the early stages of the war. This may seem strange in view of the prior concentration upon the liquid fuel type, however, the promise of a more immediate return from the studies of solid fuels resulted in intense research upon this type of propellant. By solid fuel is usually meant a gunpowder mixture, commonly a mixture of nitrocellulose and nitroglycerine. Other mixtures are of course feasible and have been used.

Liquid fuels have come into prominence since World War I, largely a result of the work of Dr. Goddard. As in the case of solid propellants, the fuel proper plus an oxy-



Moby Dick, experimental rocket propulsion unit. Only twenty inches in diameter, it produces one third more thrust than the German V-II Rocket. Photos, courtesy U. S. Navy gen source is required. After considerable research, it was Dr. Goddard's conclusion that the best type of fuels were light hydrocarbon gases. For reasons of expediency, hydrogen and gasoline have been more commonly used, while alcohol because of its energy "availability" has come into considerable prominence.

Several advantages and disadvantages attend the selection of either type of fuel. Solid fuels permit an extremely simple type of rocket. The powder in one of several forms is placed in the cylindrical "motor." Ignition is caused by a small igniter, and the gases of combustion are expanded directly through a venturi at the rear of the combustion chamber. Of particular disadvantage in this type of rocket is the inability to control the rate, or period of burning except by the original design of the rocket. For this reason, solid-fuel rockets are designed to accelerate rapidly, and then coast through the balance of the trajectory much as an artillery shell.

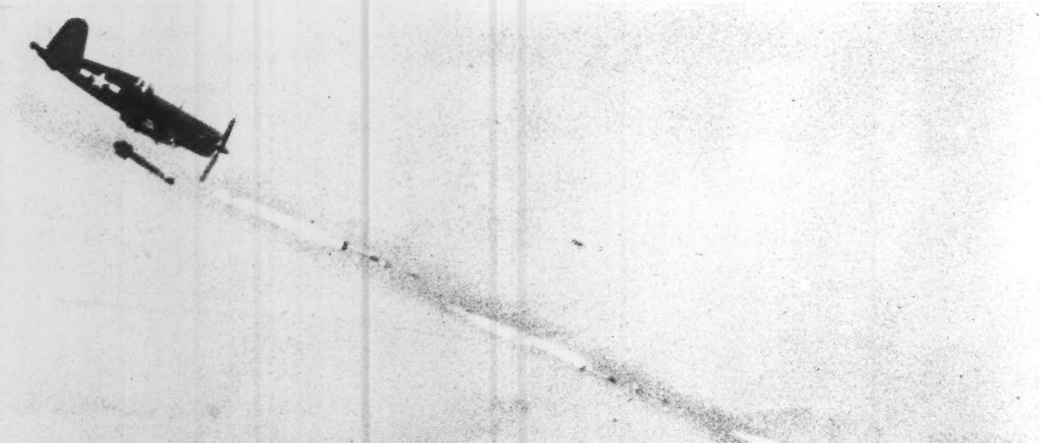
Liquid Fuels

Liquid-fuel rockets eliminate a number of the disadvantages found in the solid type. Fuel may be ignited and shut off at will permitting the rocket to be accelerated for a period, then allowed to coast. Fuel is thus economized. Trajectory or flight more similar to that of the conventional aircraft is realized. A distinct disadvantage, however, is the need for a fuel pumping system producing pressures in excess of the pressure in the combustion chamber. Two approaches have usually been made to this problem. The first, which is the simpler, is to pressurize the fuel tanks. However, maintaining the pressures in the correct proportion is not readily solved in this manner. Literally pumping the fuel to the combustion chamber has been considered a better method. The disadvantage to this system lies in its bulk and the energy requirements of the pump. Power generated through spontaneous chemical reaction has been generally used for the purpose.

Of the larger size rockets, two solid fuel propelled, one liquid propelled, all developed by the Navy Department, have recently come to public attention. A solid fuel propelled, 1000-pound rocket known as the "Tiny Tim" was developed as a means of carrying a 500-pound, semiarmor piercing "pay load." Twelve inches in diameter, and ten feet long, the rocket is propelled by the expansion of gases through a number of venturi arranged in a circle at the base of the rocket. In this type of missile, flight stability is achieved through the use of fins placed along the side of the rocket. As in all fin-stabilized projectiles, it may be surmised that the accuracy is relatively poor, and that the flight is more or less erratic. This condition has been improved in solid-fuel rockets through the use of tangential venturi, thus achieving a rotational stability similar to that obtainable with rifle-fired shells.

Indicating the wide range in rocket types possible with a solid type of fuel, is the 20-inch diameter rocket known as "Moby Dick," developed for the Navy by the Aerojet Corporation. This rocket, while only slightly over nine feet long, develops a thrust in excess of thirty tons, surpassing by more than one-third the thrust of the considerably larger V-II rocket developed by the Germans. In this motor, the gases expand through an axial venturi, indicating its use in conjunction with either fin or wing-stabilized projectiles or planes.

Pointing the way toward the commercial rocket-propelled aircraft of the future, is the "Gorgon" pilotless aircraft developed by the Navy, Bureau of Aeronautics. Propelled by a liquid-fuel rocket engine known as the "CML 2N," the compact motor delivers a thrust of 350 pounds, controllable at will for a total combustion period of over two minutes. In this rocket, the fuel—analide and nitric acid—are stored in separate chambers, and are pumped to the combustion chamber, which is only four inches in diameter and ten inches long. A system known as regeneration has been used in this design. The oxidizer passes from its tank through a space around the combustion chamber, thence to the injector. By this means, not only is the liquid given additional energy in the form of heat, but the combustion chamber is kept cool, simplifying the mechanical design and the metallurgy.



A Navy F-4U launches a volley of rockets. The large rocket below the plane is a "Tiny Tim" already dropped, and ready to ignite

What Cemented Carbides Offer the Designer

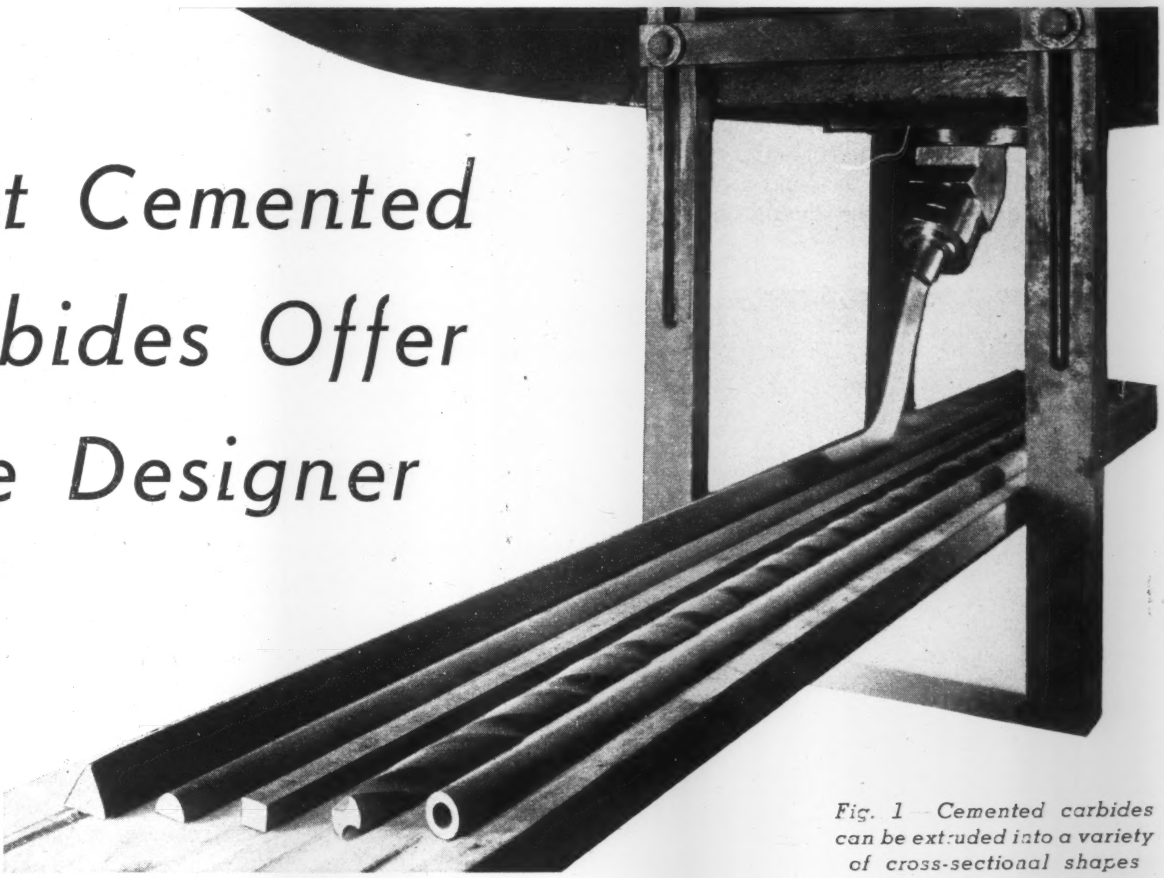


Fig. 1—Cemented carbides can be extruded into a variety of cross-sectional shapes

By Richard K. Lotz

Associate Editor, Machine Design

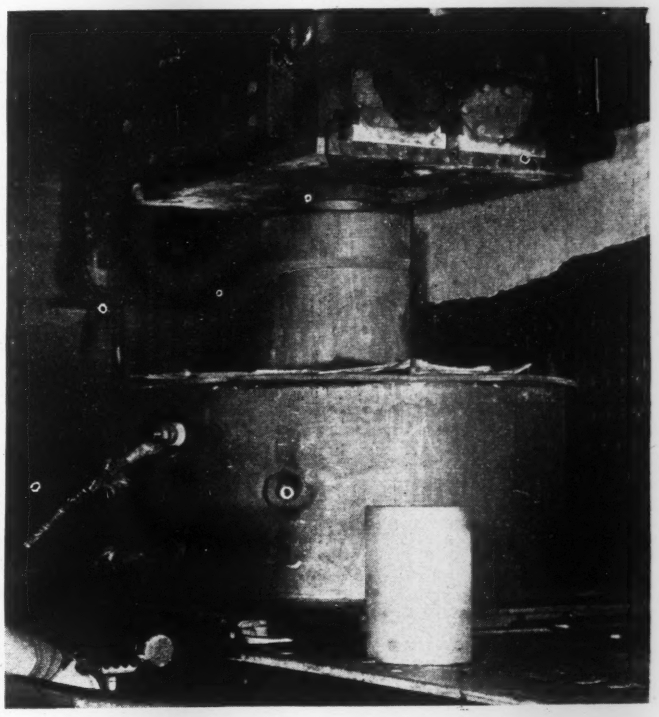
WHERE machine parts require extraordinary wear resistance, the designer does well to consider the possibility of employing one of the cemented carbides. These are available in a variety of grades, can be formed to specified shapes to close tolerances, approach the diamond in hardness (a property which is retained at extremely high temperatures), have high compressive strength, and offer good general corrosion resistance.

Strictly speaking, cemented carbides are powder metals, for it is by the process of powder metallurgy that they are produced. Powdered tungsten, tantalum, titanium, molybdenum, carbon, cobalt and nickel are employed in numerous combinations in which all or some of the first four metals are combined with carbon to form carbides, the cobalt or nickel being used as the cementing medium.

In producing cemented carbide parts the first step is making the required carbides. In the case of tungsten carbides, pure tungsten powder is mixed with pure carbon powder (lampblack) and fired at a temperature of from 2500 to 2700 F. The carbides thus produced are then mixed with the relatively soft binder material—powdered cobalt or nickel—and the resulting powder is formed into the desired shape by either cold pressing, hot pressing or extruding. Which method is used depends on various factors such as size, shape and quantity of the pieces required.

Cold pressing is employed for (1) molding parts to

Fig. 2—Below—This hydraulic press, used for hot pressing carbide parts, has capacity of 100 tons. It handles parts up to 100 sq. in. cross section with depth of 8 inches



specified size and shape and (2) molding ingots and slabs from which special shapes can be formed by ordinary machining methods. In the first case, an individual die or mold is required for each shape; after the powder has been pressed to shape, the resulting compact is presintered (fired) at a relatively low temperature. Next, it is sintered at a much higher temperature and cooled. The material acquires its extreme hardness during sintering and it is interesting to note that shrinkage of the pressed parts during sintering may run as high as forty per cent in volume. This method of making cemented carbide parts is particularly well adapted to high-quantity production.

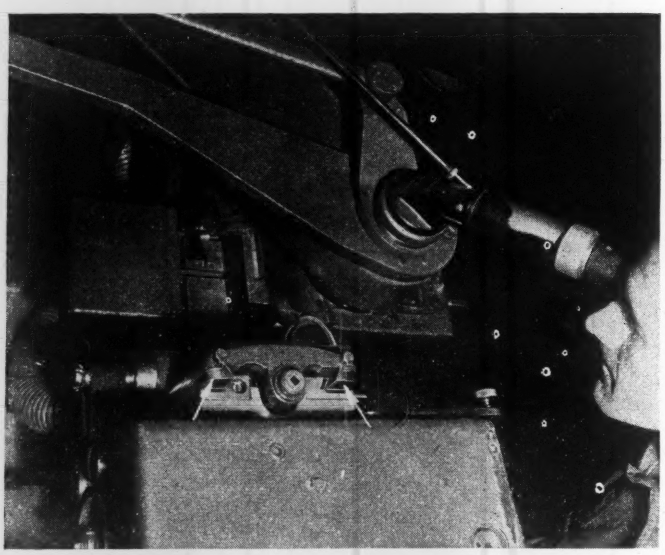


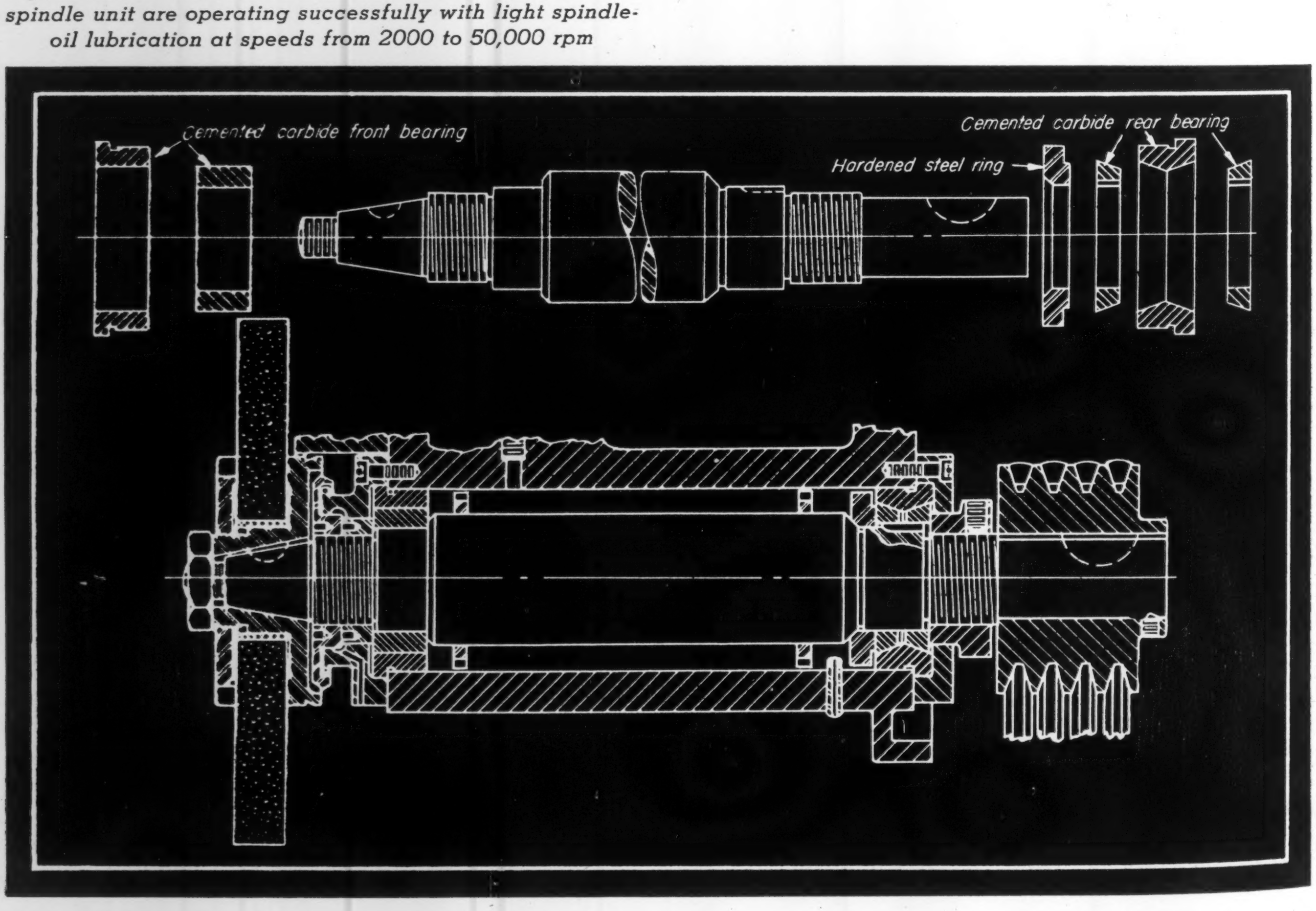
Fig. 3—Above—Arrows point to ways of profile grinder which are tungsten cemented carbide and outwear cast iron

The second manner of employing cold pressing is most suitable where limited quantities are involved or where the shape required cannot readily be produced by pressing. This procedure involves pressing the powder into ingots or slabs which are presintered to give them strength sufficient to permit handling and working them. These slabs have, roughly, the consistency of chalk—albeit they are much more abrasive—and can be machined by milling, turning, drilling, grinding, etc., to any shape desired. The parts thus made are then hardened by sintering.

Hot pressing combines in a single operation the three distinct processes of pressing, presintering and sintering. Powders are pressed in molds of a graphite base composition which employ electrical resistance heating to create the high sintering temperatures required. The hydraulic press shown in *Fig.* 2 has a capacity of 100 tons and handles the hot pressing of parts up to 100 square inches in cross section with a depth of eight inches. Hot pressing is used for parts that are too large to be sintered in available furnaces, or for thin-walled parts that tend to go out-of-round when cold pressed and sintered. Typical parts being produced by hot pressing are: Shell nosing dies for 90-mm and 105-mm shells, tapering and casing dies for the same caliber ammunition, and broaching rings for finishing aircraft engine silver-lined bearings.

In the extruding of cemented carbides, the powdered metals are mixed into a paste, extruded through shaped nozzles, dried, presintered, cut into lengths as required and, finally, sintered. As is shown in *Fig.* 1 a variety of

Fig. 4—Below—Carbide-on-carbide bearings in this grinder spindle unit are operating successfully with light spindle-oil lubrication at speeds from 2000 to 50,000 rpm



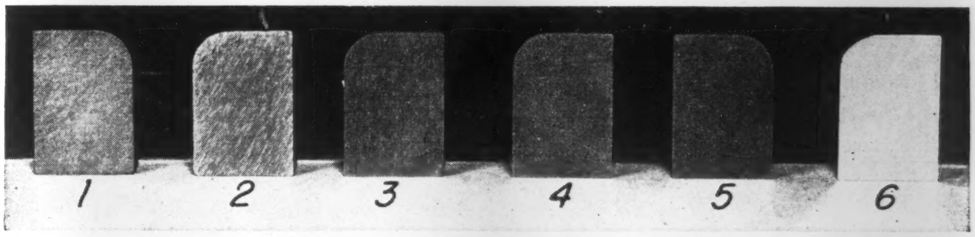


Fig. 5—Right—Samples illustrate finishes obtainable on cemented carbide part surfaces: (1) As-molded, about 50 microinches; (2) Ground with 90 grit silicon carbide wheel, 10-15 microinches; (3) Ground with 100 grit diamond wheel, 9-10 microinches; (4) Ground with 220 grit diamond wheel, 5-6 microinches; (5) Lapped finish, 3½ microinches; (6) Polished finish, 2½ microinches

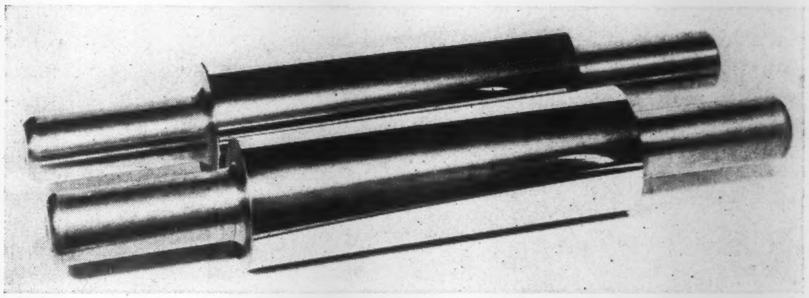


Fig. 6—Right—These solid cemented carbide rolls, used for rolling sheet steel, are approximately 15 inches overall length

cross-sectional shapes can be produced by this method.

As has been indicated, the outstanding properties of cemented carbides are: Extreme hardness, high compressive strength, high Young's modulus, and high wear resistance. Worthy of note is the fact that they retain their hardness at high temperatures, one grade being as hard at 1550 F as high carbon steel is at about 650 F and high speed steel at 825 F. Specific physical properties of some commercial grades of cemented carbides are listed in the table. While the values listed are representative of properties obtained in good production practice, they do not represent the highest obtainable nor the lowest which are of practical use.

An example of the high wear resistance of cemented carbide is found in the vertical slide ways of the Wickman profile grinder pictured in *Fig.* 3. In this application both the dovetail ways and the wear strips are tungsten cemented carbide, the dovetails measuring about 9¾ inches in length by 1 inch wide and ½-inch thick, the wear strips—inserts and gibs—measuring 3/16-inch in thickness. Originally these ways had been made of cast iron. However, under the severe abrasive conditions imposed, to maintain the accuracy required of the machine the ways had to be removed and scraped back to a plane surface every six months. The present carbide ways have been in continuous service, two eight-hour shifts per day, for more than

Physical Properties of Some Commercial Cemented Carbide Compositions[1]

Composition	Specific Gravity (g/cu cm)	Rockwell Hardness, A Scale	Transverse Rupture, (psi)	Young's Modulus[2] (psi x 10^6)	Compressive Strength[3] (psi)	Proportional Limit in Compression[4] (psi)	Impact strength[5] (ft-lb)	Endurance Limit[6] (psi)	Coefficient of Thermal Expansion[7]
97% WC, 3% Co	15.25	92.7	170,000	97.5	815,000	780,000	....	....	....
95% WC, 4.5% Co	15.05	92.3	200,000	90.5	890,000	740,000	....	....	....
94% WC, 6% Co	14.85	90 to 92	225,000	88	750,000	600,000	0.73	95,000	5.0 x 10^{-6}
91% WC, 9% Co	14.60	89.5 to 91.5	275,000	83	685,000	540,000	....	....	....
87% WC, 13% Co	14.15	87.5 to 90	300,000	80	625,000	525,000	1.10	105,000	5.9 x 10^{-6}
80% WC, 20% Co	13.55	85 to 87	350,000	73	550,000	425,000	1.75	....	....
Predominantly WC, with TaC and 13% Co	13.90	87 to 88	275,000	88	610,000	475,000	....	....	7.25 x 10^{-6}
Predominantly WC, with TaC and 6% Co	14.70	91 to 92	220,000	91	752,000	670,000	0.65	85,000	....
Prodominantly WC, larger amount of TiC, 7% Co	9.00	92 to 93	150,000	88	725,000	....	....	....	7.0 x 10^{-6}
Predominantly WC, with TaC and TiC, 8% Co	11.7	91.5 to 92.5	165,000	72	720,000	....	....	....	6.75 x 10^{-6}
Predominantly WC, with TaC and TiC, 11% Co	11.6	90.5 to 91.5	175,000	81	680,000	....	0.60	85,000	6.0 x 10^{-6}

[1] Values given in this table are representative of properties obtained in good production practice. They are not necessarily the highest obtainable, nor do they represent the lowest which are of practical use.

[2] Most values for Young's modulus were obtained by W. H. Davenport of the Norton Co., by the musical pitch method.

[3] Most values for compressive strengths are given through the courtesy of P. W. Bridgman of Harvard University.

[4] Proportional limit in compression is the load per unit area at which the increase in strain ceases to be directly proportional to the increase in stress.

[5] Impact values are from unnotched specimens of approximately ¼-in square section. Charpy machine was used.

[6] Values for endurance limits are based on 20,000,000 cycles, for specimens of R. R. Moore rotating beam type.

[7] Average coefficient of expansion per degree Cent. for the range 20 to 700 C (68 to 1290 F).

seven months and no signs of wear have been detected as yet. Oil grooves have been ground in the carbide ways and lubrication is by gravity feed from oil cups.

The foregoing application is only one of many in which the hardness and wear resistance of cemented carbide make it the logical choice over other materials. Typical of the wide variety of machine parts now being made of cemented carbide are: Pins and rollers, valve tappets, bushings, contact points, cam followers, chuck jaws, pressing dies and rams, bearings, mill rolls, spindles, work rests for centerless grinders, balls for sizing and hardness testing, drill jig bushings, needle valves, ball check valves and seats, seaming rolls, wire rope dies, guides, feeding fingers for automatic machines (such as filling, packaging, wrapping, etc.), and band saw guides.

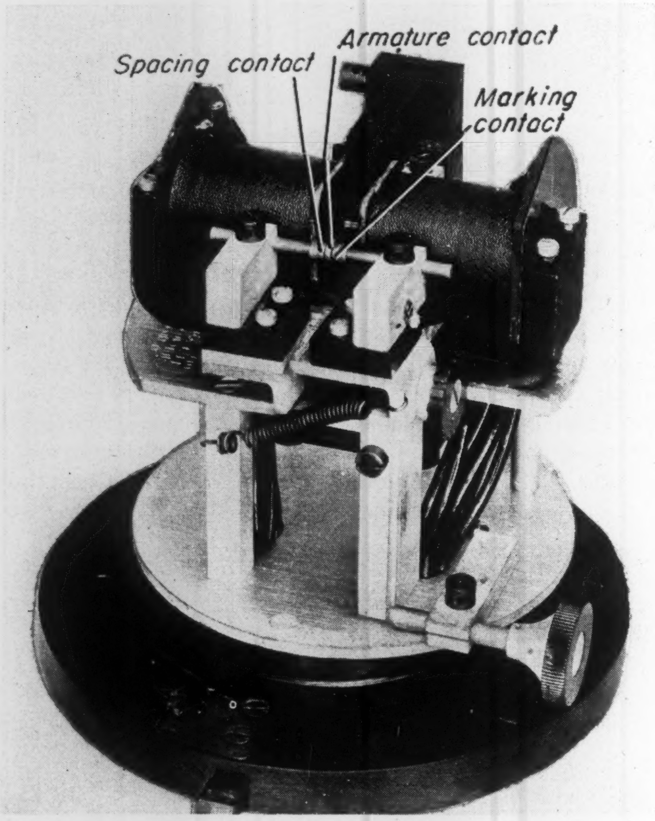
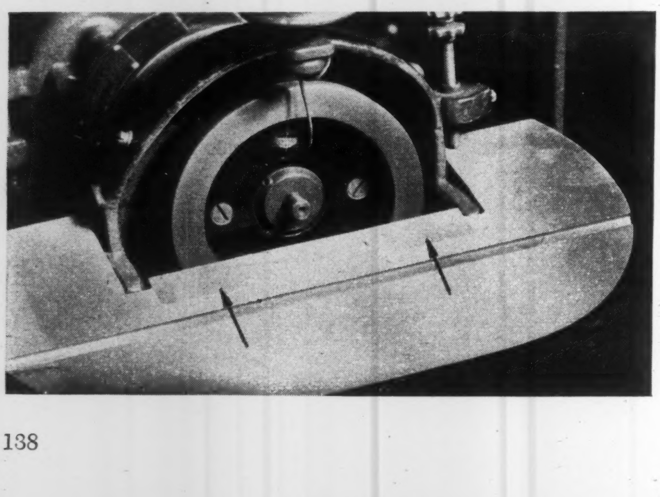


Fig. 7—Above—Carbide contacts of this relay are self-cleaning, have long life, free of servicing troubles

Fig. 8—Below—Brazed-in cemented carbide inserts curtail wear at points of severest use on this tool grinder table



Numerous low and high-speed bearings in which cemented carbide is used for both the stationary and rotating members are being operated today with marked success. Examples of this are the radial and combination radial-thrust bearings shown in *Fig.* 4. The unit pictured cross-sectionally is the wheel spindle assembly of a surface grinder. Cemented carbide bearings of this type are reported to be operating successfully with light spindle-oil lubrication at speeds ranging from 2000 to 50,000 rpm. Of course, as would be true in similar cases, surface finish has much to do with the success of the design.

Some idea of the quality of surface finishes obtainable on cemented carbide parts is given in *Fig.* 5. With the exception of part No. 1 which is shown as-molded, all other parts in the photograph have been finished by various abrasive means and the quality of the finishes is given in microinches in the caption. Further evidence of the high-luster finish that can be imparted to cemented carbide is afforded by the rolls shown in *Fig.* 6. These solid cemented carbide rolls, which are about 15 inches in overall length, are used for rolling sheet steel and, because of their resistance to deflection, high compressive strength and nongalling properties, last longer than steel rolls and produce an exceptionally fine finish on the steel sheet.

Carbide Contacts Are Self-Cleaning

In the electrical field, cemented tungsten carbide contacts are reported to be giving materially better service in modern high-speed, low-amperage telegraph transmission apparatus than conventional contact metals. The carbide contacts, attached by brazing to the marking and spacing contacts as well as to the armature of the relay shown in *Fig.* 7, are self-cleaning; that is, material which becomes loosened through arcing drops away from the carbide instead of fusing to the opposite contact as often occurs on contacts made of conventional metals. Thus the carbide contacts give longer life and do not have to be burnished or cleaned with a file.

To help curtail wear and consequent loss of accuracy in the tool grinder pictured in *Fig.* 8, cemented carbide inserts are employed in the table, being located where the tools are rubbed across the table during grinding. All that is required in such cases is the milling of pockets to receive the inserts, brazing the inserts in place and then grinding the entire surface of the member involved.

DESIGN TIPS: When designing parts which are to be made entirely of cemented carbide and mechanically held in place at assembly, it is well to keep in mind the difference in thermal expansion rates between the carbide and the material to which it is fastened. Carbides of various grades have expansion rates ranging from 5.5 x 10^{-6} to 8.0 x 10^{-6}. This is roughly half the thermal expansion rate of carbon steel, for example, which generally is considered to be 12.0 x 10^{-6}. Allowance should be made to permit relative displacement or, if the carbide part must be loaded in fastening, care should be exercised to make sure it is loaded in compression, not tension.

Use of set screws for holding carbide parts in place generally is not practical. The set screw point will not embed itself in the extremely hard carbide and so will not establish a good grip on the carbide surface. Seating the carbide part against a positive shoulder and holding it against

(*Concluded on Page* 184)

How To Avoid Loosening of Bolts and Nuts

By R. J. Sweeney
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LOOSENING of bolted fastenings in service is a problem that has caused untold damage, expense and vexation to everyone concerned with mechanical assemblies. Loosening may result from wear of the surfaces, compression of burrs, powdering of plating or scale, or similar effects. Also, it may be caused by relative rotation of the connected parts, as in a scissors joint. The surface on which the nut bears may have friction enough for the unscrewing torque but may slip when trying to apply the larger screwing-up torque. These cases are obviously bad design or bad workmanship which can be avoided.

In many cases, however, joints which are well designed and competently assembled also loosen. The failure is usually attributed to "vibration" or "shock". Now the question arises: "How can vibration loosen a tightly bolted joint? Mechanical vibration is a cyclic motion of a machine element, an assembly or a structure. If two connected parts have different natural frequencies of vibration, and are forced to vibrate together then there will exist a cyclic force on the bolted connection. When the vibration has a component along

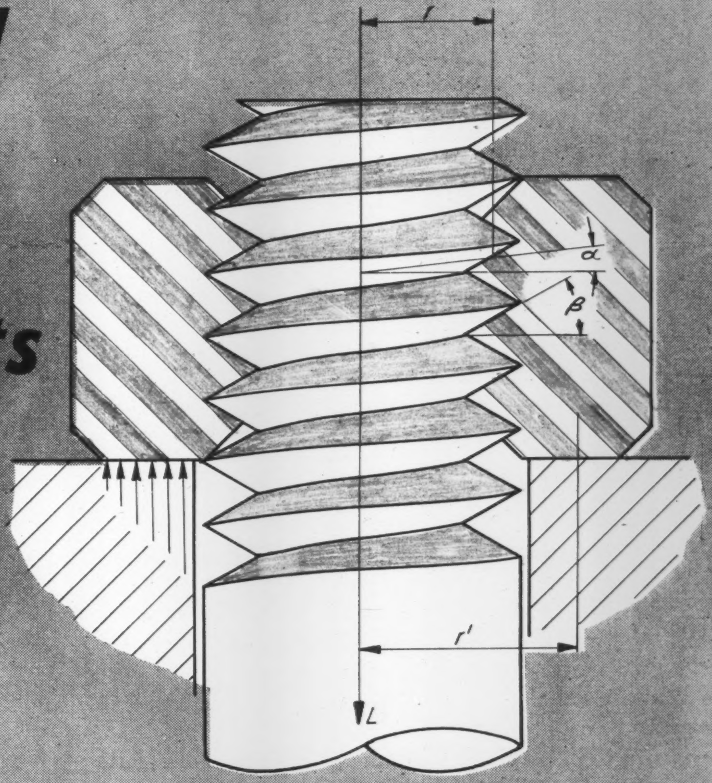
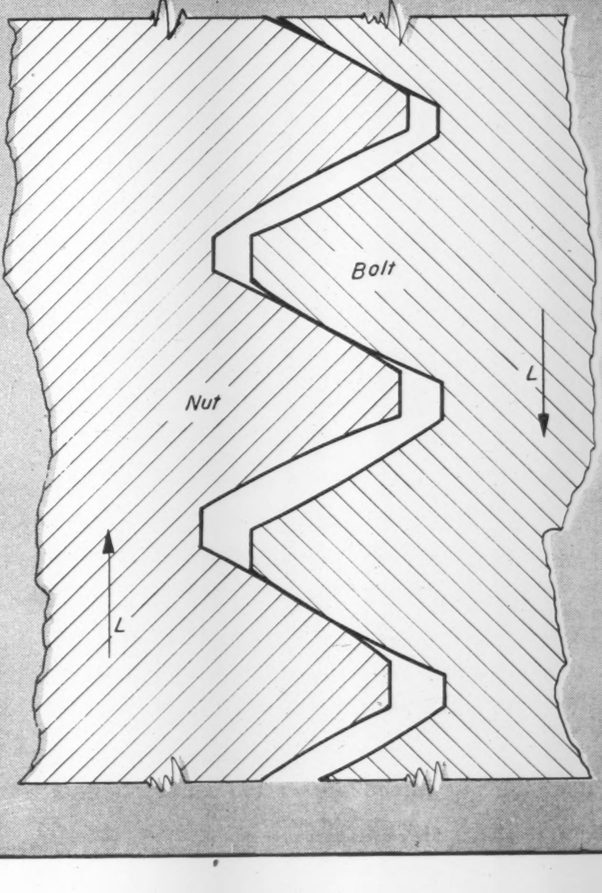


Fig. 1—Above—Nut and bolt assembly, indicating helix angle α, thread form angle β, pitch radius r, base radius r', and nonuniform pressure on base of nut

Fig. 2—Below—Section of nut and bolt threads showing cantilever bending of threads and separation due to radial component of load



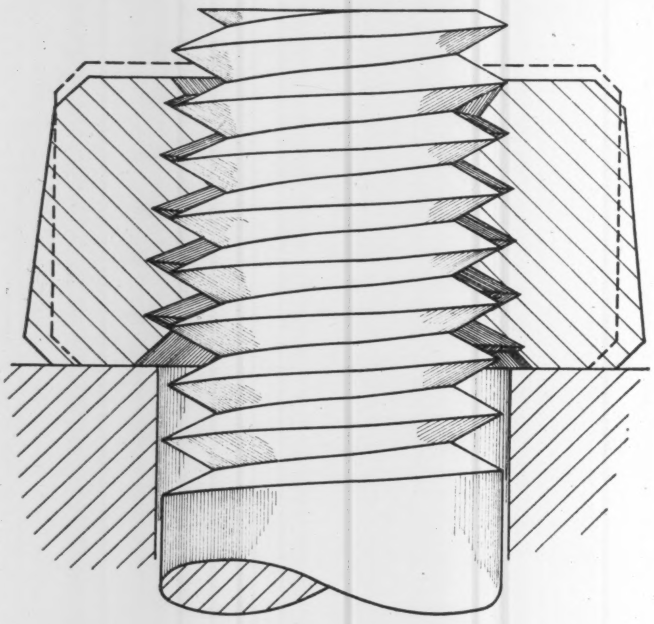


Fig. 3—Nut under load, showing the axial contraction and radial expansion concentrated at the base

the axis of the bolt, the bolt tension will fluctuate due to the cyclic force. Since this is an effect of vibration which appears in the threaded connection, it is a possible cause of loosening.

Load Produces Unscrewing Torque

Static Analysis: Reviewing the situation in a threaded fastening under static tension, as in *Fig.* 1, the bolt load produces an unscrewing torque due to the helix angle of the thread. Its magnitude is $T_0 = Lr \tan \alpha$, where L is the bolt load, r the pitch radius of the thread and α the helix angle. The torque T_0 will be called the normal pressure torque. Under static conditions this unscrewing torque is resisted by the thread friction and the friction of the base of the nut on the work. The thread friction torque will be approximately $Lr\mu/\cos\beta$, where μ is the coefficient of friction and β is half the thread form angle (30 degrees for the American Standard). The base friction torque will be $Lr'\mu$, where r' is the effective radius of the base. If these torques are worked out for the ¼–20 National Coarse thread Regular nut, the friction torques are approximately 5.7 times the normal pressure torque and for the 1–8 National Coarse thread Regular nut the factor of safety is about 8.8. Thus, under static conditions the frictions will effectively prevent rotation of the nut by the normal pressure torque.

Load transfer between the nut and bolt threads is concentrated near the base of the nut with relatively low pressures near the free end of the nut. This becomes evident when the deformations of the bolt and nut are considered under load. The pitches of the nut and bolt threads are equal in the unloaded condition. Under load, the bolt elongates, which increases the bolt thread pitch; the nut is compressed axially, which decreases the nut thread pitch. The thread bearing pressure is thus concentrated at the lower end of the nut thread during the initial stage of loading. This condition is shown by photoelastic studies of threads made by Hetenyi (1)*. The situation is modified at higher loads. The threads are deflected as cantilever beams, as in *Fig.* 2, and the nut is expanded radially, thus transferring the load up into the nut. With increase of load the thread pressure becomes more nearly uniform throughout its length.

Goodier (2) has shown this effect by a different method. He used a bolt with a single-turn thread and measured the deformation of the outside of the nut with the single bolt thread placed at varied positions in the nut. With the thread at the bottom of the nut, the deformation of the nut was closest to the shape assumed with a complete thread on the bolt.

Action of Nut Under Load

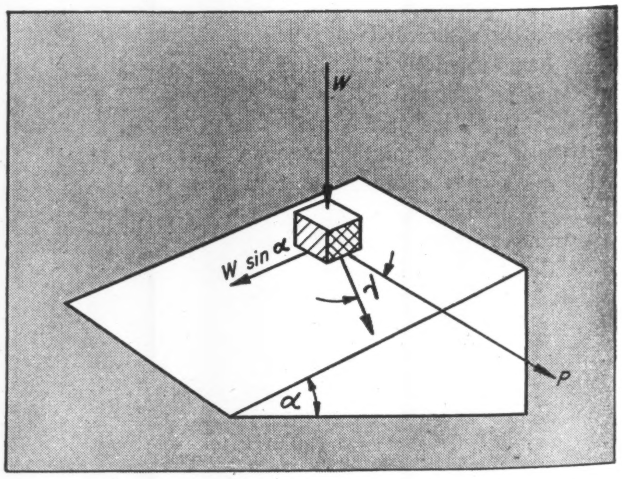
Motion Under Load: The displacements of a nut and bolt under load are complex and not susceptible to easy elastic analysis.

1. The nut wall expands radially under the action of the radial component of bolt load due to the thread form angle β and also because of the Poisson effect of the axial compression. This displacement is greater at the base of the nut than at the free end, ~~as indicated in~~ *Fig. 3.*
2. Similarly the bolt contracts radially because of the same forces.
3. The nut is distorted by the shear loading due to the downward pull of the threads and the upward reaction of the base acting on different radii. This will concentrate the base reaction near the inner radius of an initially flat base, thus reducing the value of the base friction radius r'.
4. The normal pressure torque will twist the bolt and nut, both in the unscrewing sense.

With decreasing load all of these motions are reversed.

Under dynamic load variation the nut threads are moving radially relative to the bolt threads, and the nut base is moving radially relative to its seat. The thread and base friction must resist this motion and the direction of the friction forces is radial unless the nut is turning. The normal pressure torque tries to unscrew the nut and must be resisted by a circumferential component of friction. The

Fig. 4—Block sliding on inclined plane under the influence of gravity and a transverse force



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*Numbers in parentheses refer to references at end of article.

Comparing the Merits of Front and Rear engine autos

By **Austin M. Wolf**
Automotive Consultant
New York City

Part II

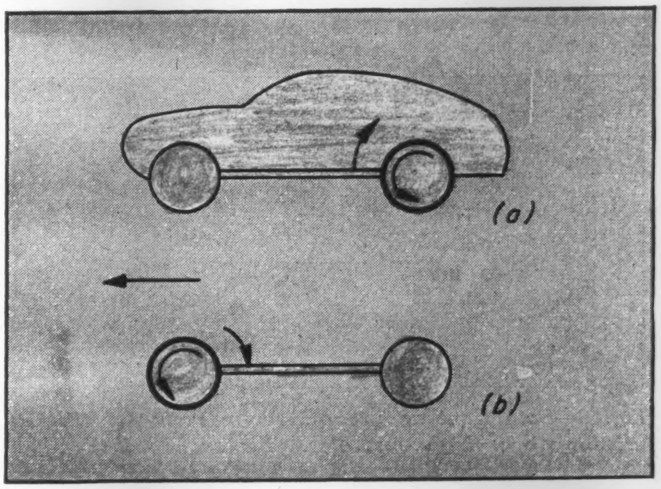
AS WAS indicated in last month's article, both front and rear-engined autos have their advantages and drawbacks. Such being the case, it becomes apparent that the most advisable procedure is to discuss impartially the factors involved and let the reader draw his own conclusions.

INFLUENCE OF TORQUE REACTION: With forward motion, particularly in accelerating a vehicle with rear-drive, the torque reaction tends to raise the front end of the chassis and cause "rearing". This will be recalled as an occasional hazard with the original Fordson tractor when not operated properly or on an upslope. This rearing tendency is portrayed diagrammatically in *Fig.* 7 *a*. On the front-driven car, the torque reaction is, of course, still clockwise and tends to depress the rear end of the chassis, but the rear tires act as an elastic medium or shock absorber to snub this moment, *Fig.* 7 *b*. Since deceleration rates are far in excess of acceleration rates, the critical condition occurs when braking, with nosing-down.

This would be counteracted by the rear weight of the power plant shown in *Fig.* 1 *c*[*] but under acceleration rearing proclivities are present with this construction. It could be partly overcome by placing the engine to the front of the rear wheel center, instead of to the rear as shown in *Fig.* 4[*], but it would encroach too much on the passenger space, thus ruling it out. This layout has been used, however, in a few racing cars. In the Scarab, the engine is located above the frame-mounted differential, as shown in *Fig.* 8, with a vertical drive at the front from the clutch to the transmission and then to the differential below the engine. Short transverse propeller shafts extend to each wheel.

OVERCOMING OBSTRUCTIONS: Under ordinary level road conditions, the tractive factor would be the same whether the vehicle is front or rear-driven. (When it comes to abnormal obstructions, the front-driven vehicle has a theoretical momentary advantage.) Referring to *Fig.* 9, it will be assumed that there is an obstruction ahead of the front wheel, extending up approximately to the center of the wheel. With rear drive, the vehicle is absolutely blocked. With front drive, assuming that the driving tire can get a grip on the obstacle, it would go over it. A driving wheel "climbs" as well as causing translation of the vehicle, due to the ability of the pneumatic tire to grip by conforming to contour. The figure shows an exaggerated case but, taking an ordinary set of steps, it is apparent that a vehicle with front drive could initiate movement up the steps

Fig. 7—Torque reaction during forward acceleration causes "rearing" of (a) rear-driven and (b) front-driven cars



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[*] "Comparing the Merits of Front and Rear Engine Autos", Part I, MACHINE DESIGN, May, 1946.

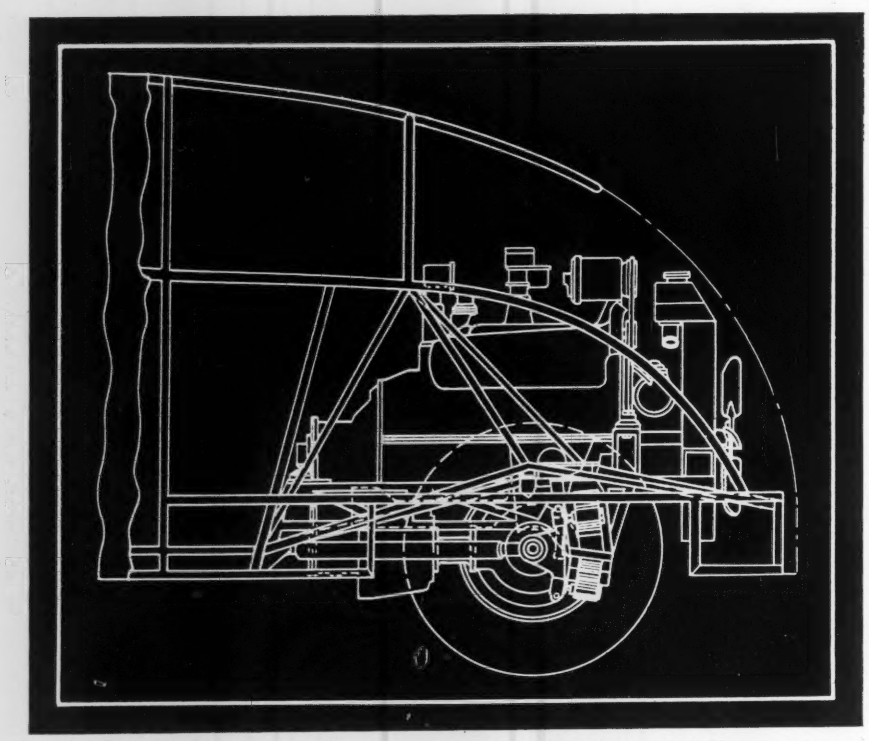


Fig. 8—In rear-driven Scarab car, engine is mounted above differential and vertical drive is employed from the clutch to the transmission

whereas the rear-driven one would have difficulty. However, this comparison is incomplete since, assuming that the front-driven vehicle in *Fig.* 9 has gone over the obstacle and furthermore that it has not become hung-up there remains the problem of pulling the rear wheels over the same obstacle, the car being blocked as in the original case with the rear-driven vehicle. The foregoing discussion indicates wherein the four-wheel drive principle is desirable for "off-the-road" service since each wheel is then able to do its "climbing" on encountering an obstacle. As for this climbing ability, it would be desirable to have the center of gravity closer to the end of the vehicle where the two driving wheels are located so that the other two wheels can be more easily pushed or pulled over the obstacle. However, this is but one phase in the whole gamut of conditions.

Driver's Position—Visibility and Protection: One of the greatest complaints with today's car, which would be increased if the power plant were projected forward causing the hood to further obstruct vision, is the poor visibility, the driver being unable to see objects along the curb or directly in front of him. With the rear-engined car the driver can be placed well forward and, with no hood and with fenders closer to the operator, visibility would reach the utopian stage.

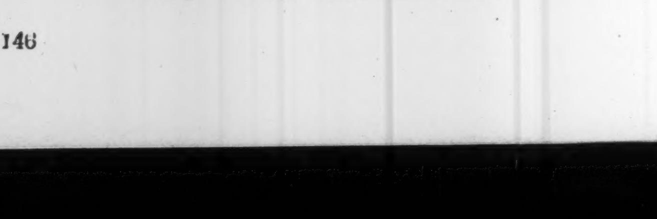


Fig. 9—Both front and rear-driven cars have difficulty climbing over high obstacles. For such "off-the-road" driving the four-wheel drive is best

Many contend that, with the driver further forward, he would be in a precarious position in the event of a head-on collision. The forwardly located power plant and the frame rails of the present day car do provide protection in the form of a large mass plus struts which, under ordinary accidents, absorb a considerable amount of the energy in a collision. However, with a violent encounter, engines have come through the dash and caused considerable damage. With a rear-engined job, most vehicles have a front storage compartment which acts as a protecting element. There is no reason why it cannot be well braced and built on the principle of a football helmet to form a crash-wall. The essential space requirement to transport passengers from one place to another is to provide sufficient room for them in comfort, and the body dimensions are determined by the space required ahead of the front seat, the space between the front and rear seats, the depth of the seats fore and aft and the termination of the body proper back of the rear seat. In *Fig.* 10 there is a central passenger space and the forward appendage can hold either the power plant or luggage and the same can be said of the rear appendage. Possibly the front compartment, when housing the power plant, might be slightly longer than the rear one as indicated by the dotted lines. The general lines of the car for streamlining would, of course, obviate the abrupt contour shown in the figure.

There is a limit to which the front seat can be moved forward. Leg and foot room must be available for its occupants. Clearance under the fenders and toward the center of the car must be provided to permit the front wheels to swing to their maximum turning position in both directions. Thus there is a definite area into which the toe and floor board zone cannot intrude in providing room for the seating of three passengers on the front seat. This zone width is limited to the space between the front fender aprons. The space available would be insufficient for two people were the floor boards to extend between the aprons and, furthermore, there would be no ingress to the seats except from the front. An aisle between them for rear entry would require spacing them still farther apart. Thus a well forward position can only be provided for one person, the operator. Motoring has a sociability aspect that would not permit such isolation.

In connection with streamlining, especially of the so-called tear-drop design, the rear location of the power plant lends itself admirably. However, streamlining in an airplane is quite different from that in an automobile since the plane always heads into the wind. An automobile can be unstable in a side wind and this has been the case with many streamline designs especially in passing a

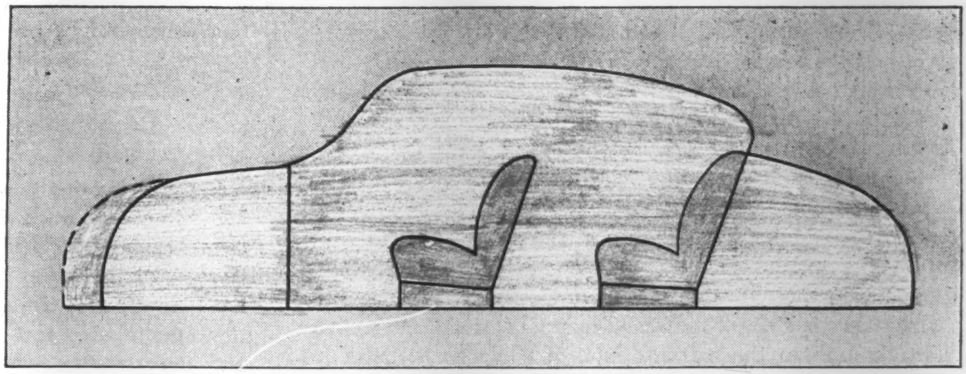


Fig. 10—Body design for either front or rear drive. Forward or rear appendage might be used to house engine, the remaining appendage being used for luggage

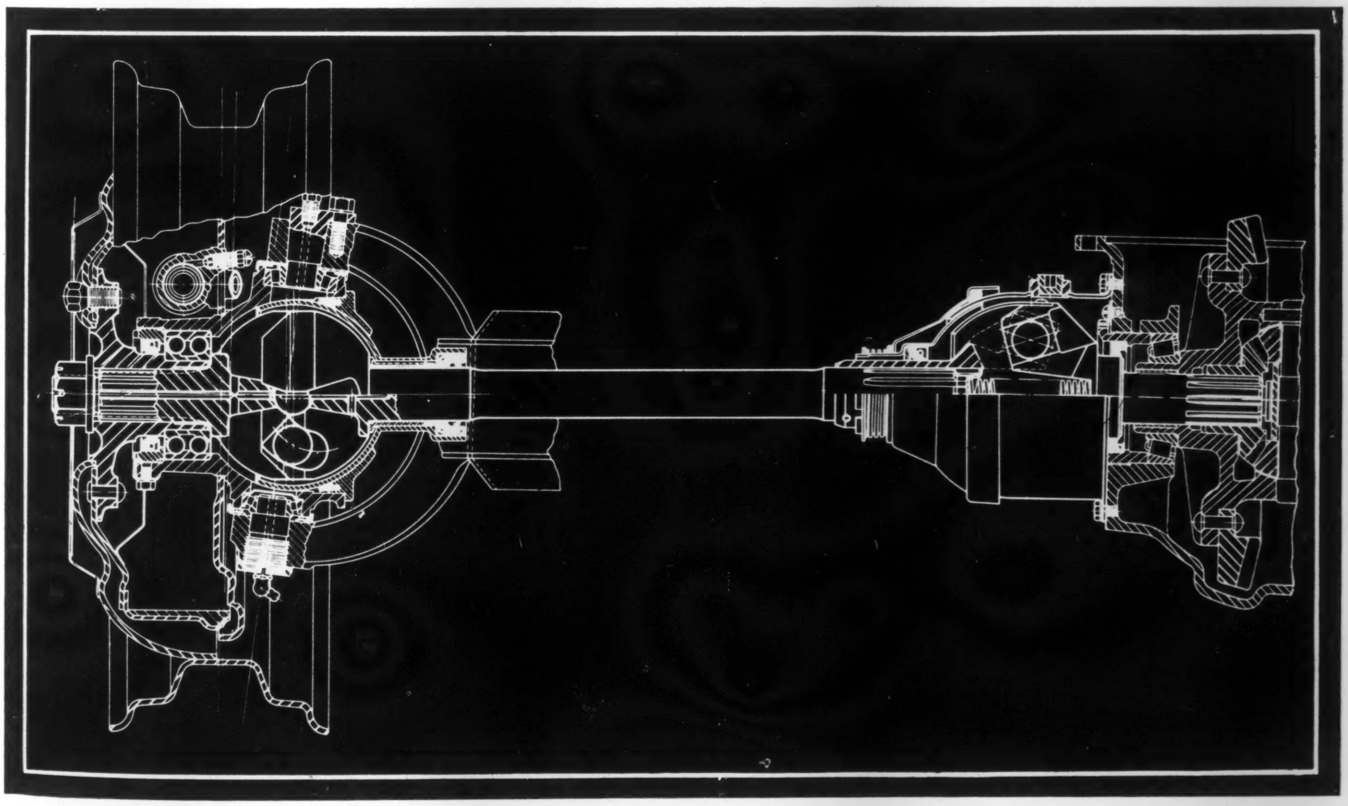
building which momentarily shields the car, after which it is exposed to the full force of the wind. The theoretically fully streamlined car is not practical for this reason and, furthermore, its length is excessive. However, its partial incorporation is desirable since wind noises are eliminated at high speed so that a conversation can be carried on by the occupants of the car which would otherwise be drowned out. Furthermore, at speeds above 40 mph, economy is attained due to lessened wind resistance.

STEERING: With a front-driven vehicle, the steering function must be added to that of propulsion. This is simple where short transverse universally jointed shafts extend between the differential and the wheels. The outer universal center lies on the steering knuckle axis and permits continuous deliverance of power to the wheel whether it is in the straight ahead position or at any angle. The Cord design is shown in *Fig.* 11, incorporating the Bendix-Weiss constant-velocity joint. The inner joint provides for end motion and eliminates the customary sliding spline. With this type of construction, there is no driving torque reaction on the wheel supporting arms—only braking reaction.

With a rigid axle housing, propulsion and braking reactions must be absorbed. This is the case in *Fig.* 3* where the conventional semielliptic springs perform this function, supplemented by a quarter-elliptic spring clamped to a center seat welded to the axle tube.

In a rigid driving axle, steering heads must be added and the construction is naturally more complicated than the conventional rear axle. A section through the Jeep steering head is shown in *Fig.* 12. This incorporates a constant-velocity universal joint of the Tracta type. Knuckle pins, joint sealing member and wheel bearings are not shown. The axle tube at the right is a press-fit on, and welded to the steering head. The right spline enters the differential while the wheel driving hub is mounted on the left spline, at the other side of the joint. As will be recalled, the Tracta joint, of French origin and manufactured by the New Process Gear Corp., consists of four

Fig. 11—Drive from differential (right) to front wheel in Cord car employs Bendix-Weiss constant-velocity joint



parts: The right and left-hand shafts, the male (or spigot) joint and the female (or slotted) joint. In the Jeep, it has a working angle of 32 degrees. The Jeeps were also turned out with the Bendix-Weiss joint until production capacity limitation forced another source of supply.

Some foreign cars utilize within their heads a Cardan type, plain bearing, double universal joint, closely coupled and with the adjacent pin axes in the same plane to neutralize the angular velocity variations that otherwise would be incurred through the use of a single joint. With a front drive, there is a slight increased manual steering effort near and through to the extreme lock of the wheels, as compared to a dead axle. However, the added effort is slight and practically nil when joints of the Weiss or Rzeppa type are used. Were a single Cardan Joint to be used in the steering head, the velocity variations would be felt back through the steering mechanism near and at extreme lock.

Engine Arrangement: The front-engined front-driven car usually has the engine crankshaft center in the conventional longitudinal position. There is available the space between the enclosures below the fenders and in any unconventional arrangement, the engine would still have to clear the wheels when they are turned to the extreme lock in either direction. In the rear-engine layout there is considerably more latitude since, without the clearance needed for steerable wheels, there is space extending the width of the body and it is possible to provide practically any desired engine position, whether longitudinal or transverse, depending on transmission and drive details.

Controls and Instruments: A front-engined front-driven car remains practically as simple as the conventional car unless, as in the case of the Cord, the forward location of the transmission would call for its remote control. With the rear-engined vehicle, all the controls must be remote including throttle, clutch and gear shift. Instruments such as the ammeter, ignition switch, oil gage and speedometer would be wired, piped and cable-connected to the rear. Heating the rear compartment would be simple but the front compartment and defrosting would call for a better installation than at present. In a long vehicle, as in the case of a bus, the weaving of the frame and/or resilience of the individual elements in the shifting mechanism might bring up problems when the control is mechanical. To obviate this, most busses have air control for clutch actuation and gear shifting.

Suspension and Drive: Most front suspension systems have less deflection than in the rear in order to prevent abnormal steering geometry under extreme compression and rebound. Such limited motion fits in well for transverse driving shafts in order to minimize joint angularity. In view of the stiffer front springs when a rigid axle housing is used, they are readily able to take the torque reaction as compared to a softer rear spring which would "wind up" to a greater extent. Such "wind-up" is inherent in the Hotchkiss drive and gives the cushioning effect.

Without the incorporation of a solid axle housing where the actual propulsion of the vehicle is transmitted through the springs (or a radius rod in some trucks), means must be provided to transmit the wheel push to the frame as is the case in the construction shown in *Fig.* 4* which merely shows the wheels in space. This function is sometimes absorbed by the suspension system, and can be accomplished with cross springs or wishbones. A swinging arm can be used, as incorporated in the Kaiser and Cord. Rigid or slightly flexible radius rods are sometimes used. The Scarab, *Fig.* 8, uses rigid radius rods. With swinging axles, the inner axle tube end terminates in a substantially horizontal fork whose axis goes through the universal joint center. This is used by Mercedes and Auto-Union. A spring steel plate, with the major axis vertical, is secured to the outer end of the axle tube in the Volkswagen and is anchored to the frame at the forward end by a rubber bushing.

Resiliency in the Drive: A disadvantage of the closely coupled power plant and transmission units in classifications 1 *b* and 1 *c** is the absence of the usual "wind up" of a long propeller shaft which gives a resilient connection between the transmission and the driving axle. The cushioning ability thus lost should be compensated for and sometimes is accomplished through the resilent mounting of the power plant.

Accessibility: If the entire power plant, transmission and driving mechanism are to be removed from the car, then it is an easier matter when all this mechanism is concentrated into one unit at either end of the vehicle. In

(*Concluded on Page* 186)

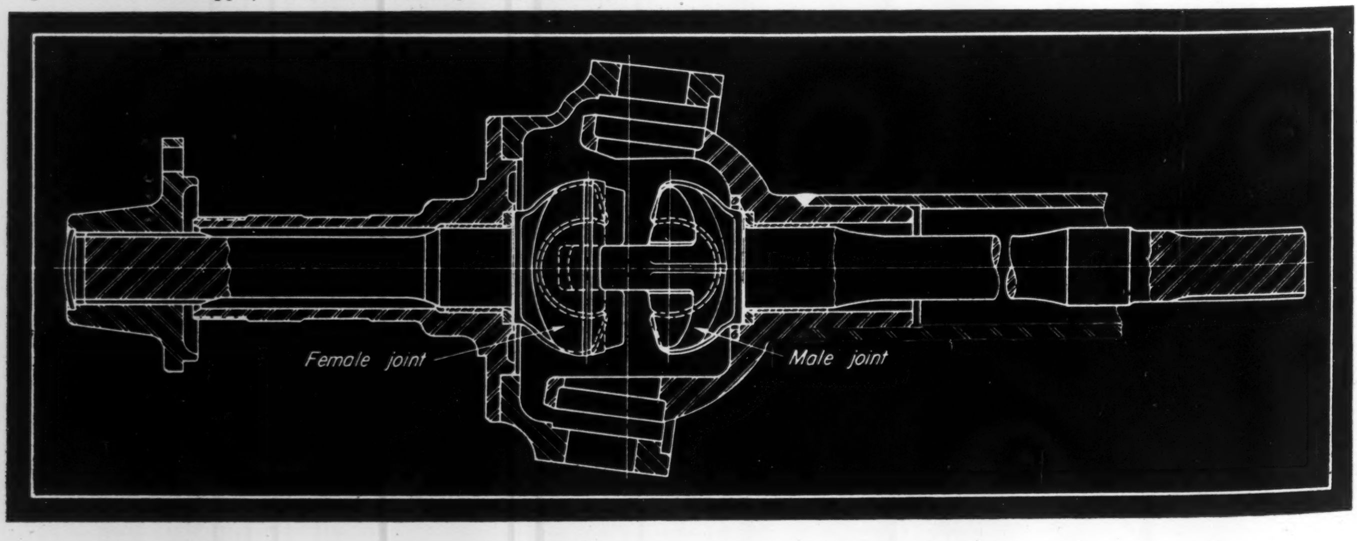


Fig. 12—Cross section through Jeep steering head, showing the Tracta type, constant-velocity universal joint

MACHINE *Editorial* DESIGN

Is Atomic Energy the Answer?

ABSOLUTE dependence of our national economy on coal as the primary source of power could not have been more dramatically demonstrated than it was during the recent strike. It is almost incredible but true that a single labor leader, during that time, brought the country to the brink of paralysis through his control over the activities of some 400,000 mine workers. Yet it must be admitted, calamitous as the results of the strike might have been, that this man rendered a service in opening the eyes of the nation to the need for a better balanced utilization of our resources and the necessity for increased intensive research and development of other sources of power.

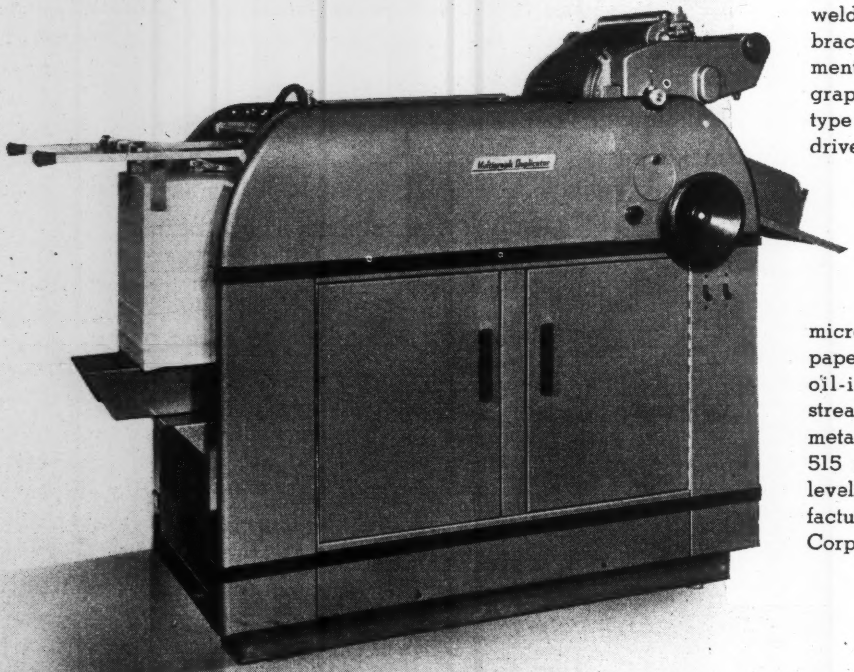
Even if the country were not faced with the possibility of further labor unrest or other disturbances restricting the supply of coal, oil or natural gas, there still remains in the more distant future the inevitable dwindling of these resources. It can readily be understood that depletion of them—particularly coal—could bring our industrial economy to a standstill just as effectively, if not as fast, as the coal strike might have done had it been allowed to continue.

Confronted with these eventualities, it would be reassuring to know definitely that research work on the development of atomic energy for peacetime uses will soon be carried forward aggressively. As pointed out by Dr. Enrico Fermi at the recent George Westinghouse Centennial Forum, it has been found that the energy releasable by fission of a ton of uranium is roughly equivalent to that in three million tons of coal. Although not all of this energy can now be used, the ratio of useful energies is still enormous. Dr. Fermi forecasts that within twenty years or so we may have large central installations in which great amounts of power will be produced and transformed into electrical energy or steam for local power consumption. Such plants might also produce plutonium which would be distributed to smaller installations in which this material, rather than uranium, would be used as the primary fuel.

While the development of atomic power for peacetime applications should continue, the research work necessary, particularly in the initial stages, cannot be separated from that connected with wartime uses. It is, therefore, hoped that a high-caliber international control can be established which will permit rapid progress to be made and which will at the same time allay any suspicion of ulterior motives on the part of all nations.

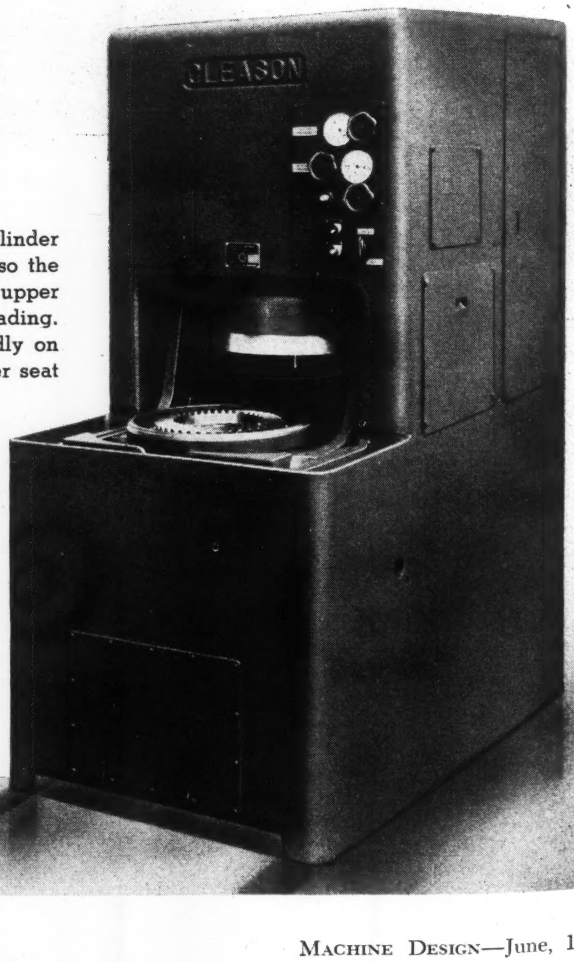
L. E. Jermy

Outstanding Designs



Direct-Process Duplicator

Featuring unit construction of welded angle-iron frame, cross-braced to insure permanent alignment of all elements, this new Multigraph has an automatic vacuum-type paper feeder, built-in power drive with centrifugal clutch, and a variable-speed control with range of 3000 to 6000 revolutions per hour. Also included are ink fountain control, an ink-roller cleaning device, fast ribbon feed, micrometer platen adjustment, wide paper-stop fingers, and needle and oil-impregnated bearings. Of streamlined design with hammered metallic finish, the machine weighs 515 pounds and is equipped with leveling feet at each corner. Manufacturer: Addressograph-Multigraph Corp., Cleveland.

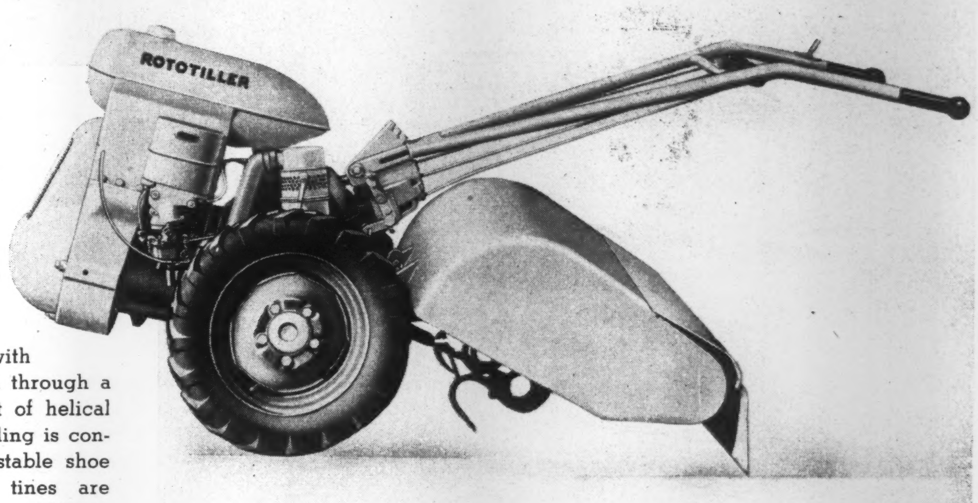


Quenching Press

Upright portion of this quenching press mounts a cylinder and piston for moving the upper die vertically and also the mechanism for swinging the lower die out from under the upper die to the front of the machine for convenience in loading. Lower die, when in position for quenching, rests solidly on the machine base and has a hardened and ground taper seat for accurately centering the lower die with the upper die. A centrifugal pump mounted to the rear of the upright with a maximum capacity of about 210 gallons of oil per minute, is so connected through valves that various rates of quenching-media flow can be delivered to the gear being quenched. Three automatic timers that work in sequence are inter-related with the above-mentioned valves, making it possible for the operator to pre-set both time and flow. Tank in base of machine holds approximately 250 gallons of the quenching media employed. Manufacturer: Gleason Works, Rochester, N. Y.

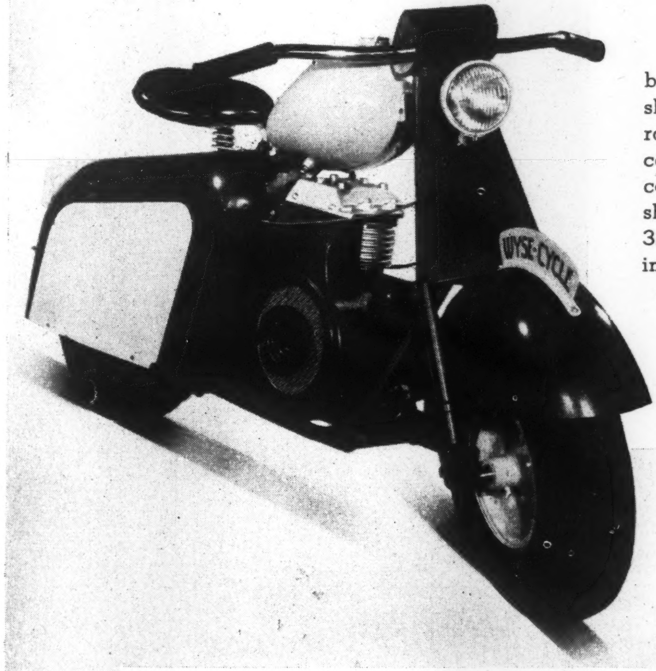
Rotary Tiller

Combining the functions of plow, disk and harrow, this new farm machine is powered by a single-cylinder air-cooled engine with transmission driven through a single-reduction set of helical gears. Depth of tilling is controlled by an adjustable shoe and the rotating tines are readily removable. Manufacturer: Graham-Paige Motors Corp., Willow Run, Mich.



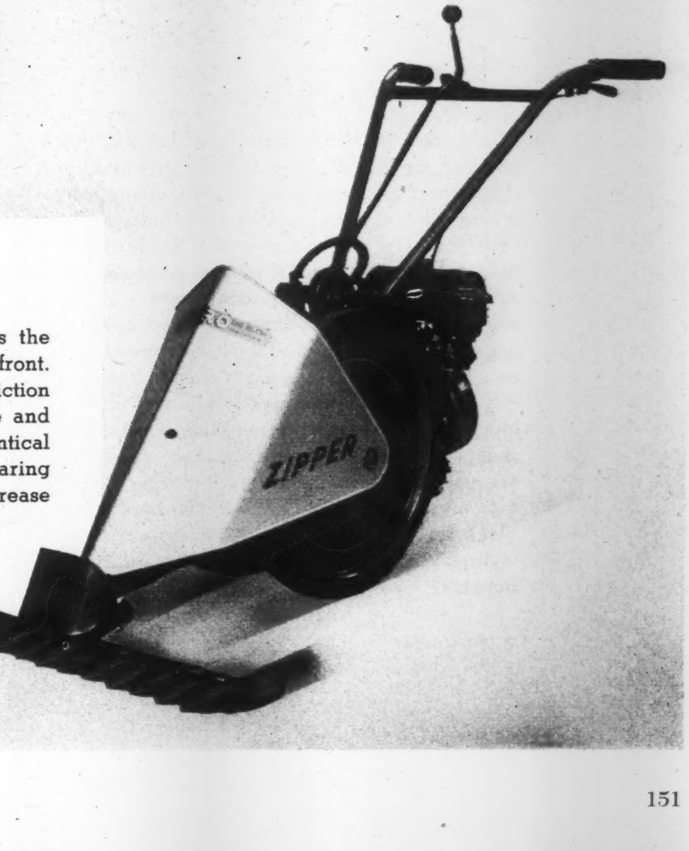
Motorcycle

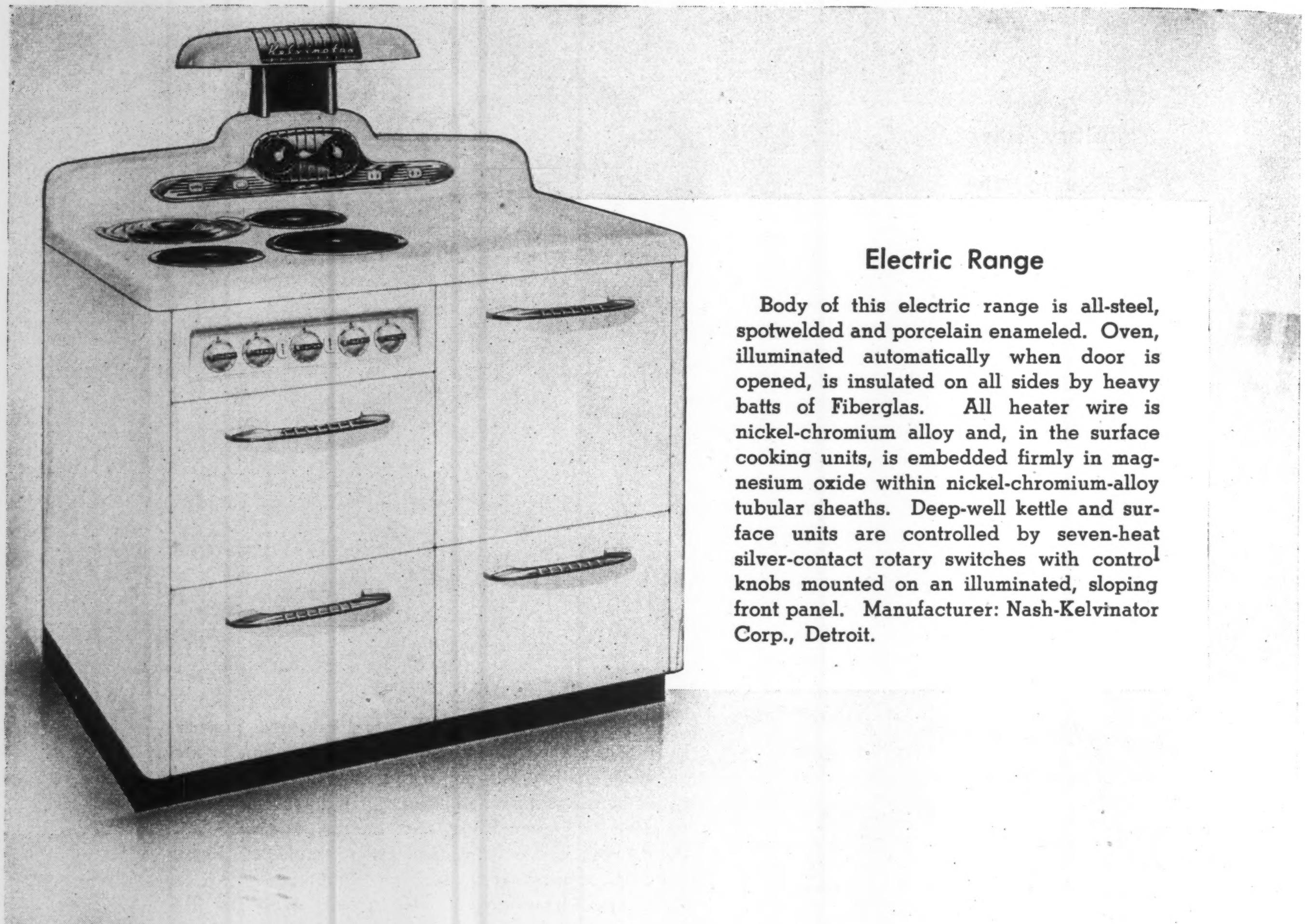
With chassis of high-alloy welded steel tubing, brazed at all joints, and body of 18-gage bonderized sheet steel, this new small motorcycle uses tapered roller bearings for wheel mountings, steering column, countershaft assemblies and in the engine. The centrifugal hydraulic transmission employed makes shifting unnecessary. Engine develops 4 hp at 3200 rpm, has bore of 3$\frac{1}{16}$ inches, stroke of 2$\frac{1}{2}$ inches, is air cooled and uses a splash-type oiling system. Capable of speeds up to 40 mph and traveling a minimum of 50 miles per gallon, this motorcycle weighs 245 lb. and has 48-inch wheelbase. Manufacturer: Wyse Laboratories, Dayton, O.



Sickle Bar Mower

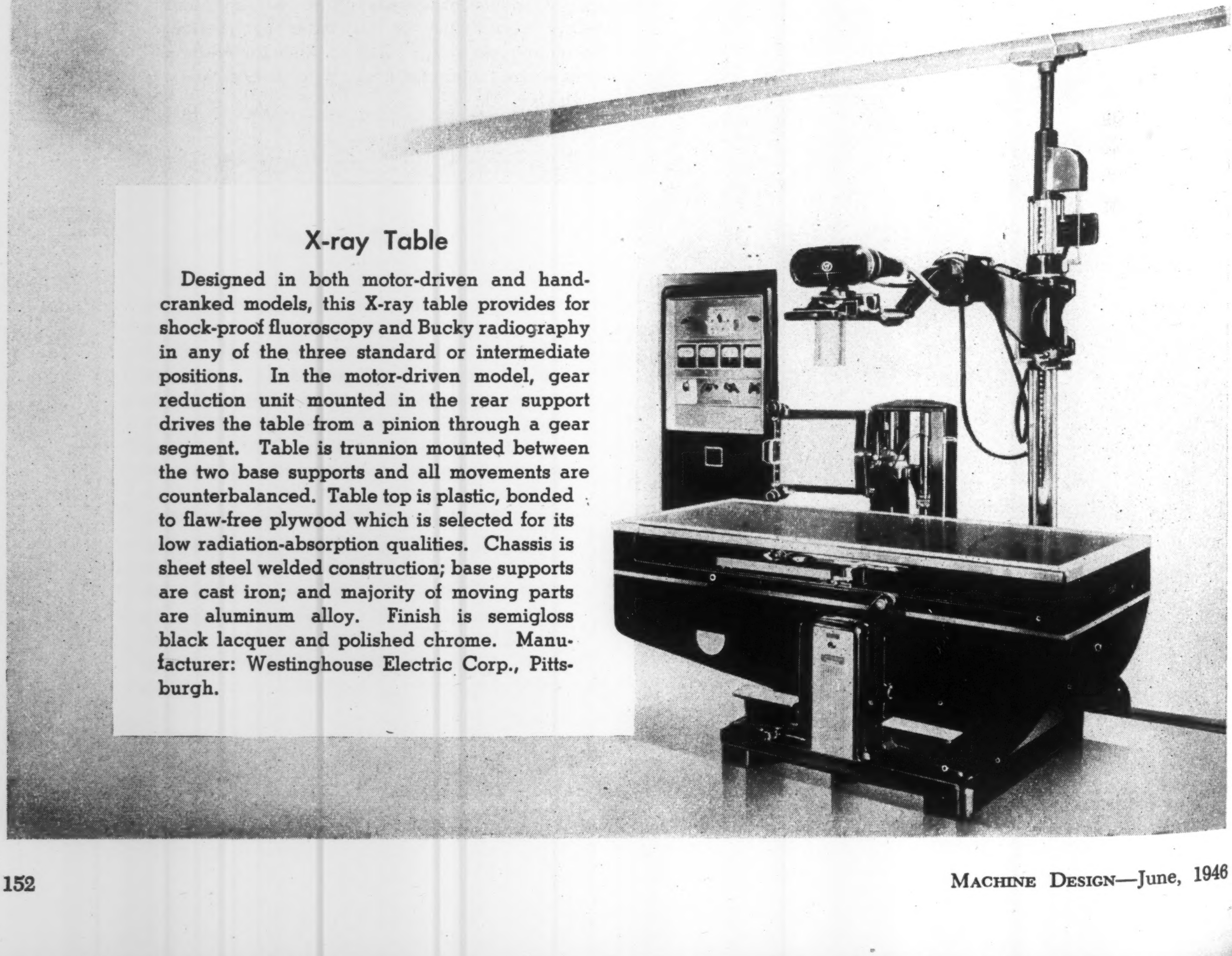
Tubular welded frame of this mower carries the engine at its rear end and the sickle bar in front. Sickle knife is shaft-driven through a safety friction coupling at the engine end and a scotch yoke and pitman wheel at the knife end. Wheels are of identical stamped disks, spot welded together with ball bearing hubs between the disks. Wheel bearings are grease packed and pitman-wheel bearing is sealed type. Manufacturer: Toro Mfg. Corp., Minneapolis.





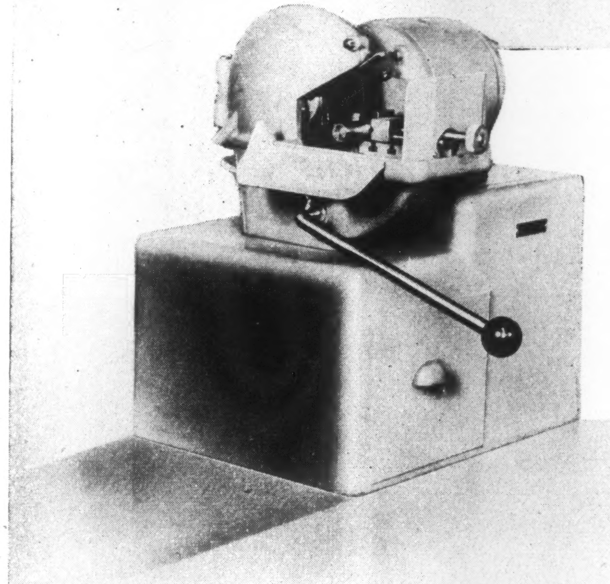
Electric Range

Body of this electric range is all-steel, spotwelded and porcelain enameled. Oven, illuminated automatically when door is opened, is insulated on all sides by heavy batts of Fiberglas. All heater wire is nickel-chromium alloy and, in the surface cooking units, is embedded firmly in magnesium oxide within nickel-chromium-alloy tubular sheaths. Deep-well kettle and surface units are controlled by seven-heat silver-contact rotary switches with control knobs mounted on an illuminated, sloping front panel. Manufacturer: Nash-Kelvinator Corp., Detroit.



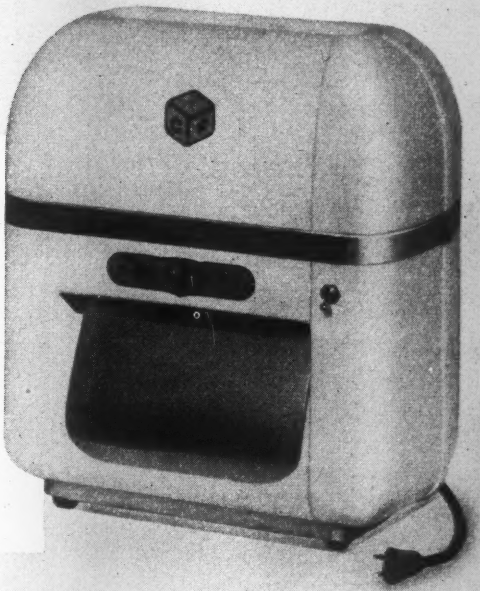
X-ray Table

Designed in both motor-driven and hand-cranked models, this X-ray table provides for shock-proof fluoroscopy and Bucky radiography in any of the three standard or intermediate positions. In the motor-driven model, gear reduction unit mounted in the rear support drives the table from a pinion through a gear segment. Table is trunnion mounted between the two base supports and all movements are counterbalanced. Table top is plastic, bonded to flaw-free plywood which is selected for its low radiation-absorption qualities. Chassis is sheet steel welded construction; base supports are cast iron; and majority of moving parts are aluminum alloy. Finish is semigloss black lacquer and polished chrome. Manufacturer: Westinghouse Electric Corp., Pittsburgh.



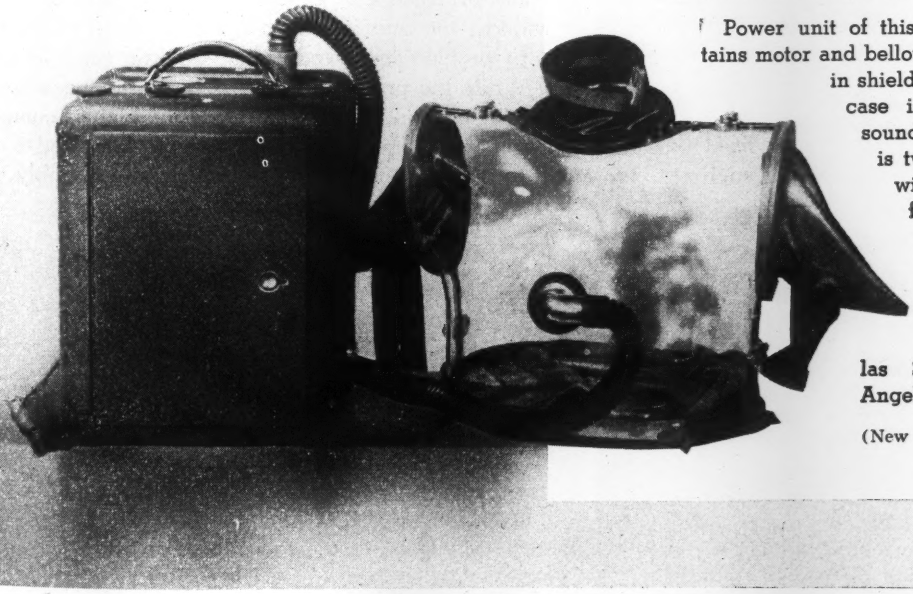
Abrasive Cutoff Machine

Base of this water-cooled abrasive cutter is welded steel finished in baked synthetic enamel. Housing and wheel guard are cast aluminum with coolant supply tubes cored in. Carriage is attached to a stainless steel rod which, mounted in brass sleeve bearings, serves as the axis of rotation and feed handle. Chrome-plated steel arbor attaches the abrasive wheel direct to the motor. Manufacturer: Buehler Ltd., Chicago.



Steak Cubing Machine

Steaks, dropped in slot at top of this machine, feed by gravity past set of knives that cube-cut them. Knife assembly can be removed for cleaning by means of a removable bearing. Outside cover is two permanent-mold aluminum castings and knife blades are high-carbon hardened steel. Power is supplied by ¼-hp motor at 1725 rpm. White baked enamel outer finish assures good appearance and easy cleaning. Manufacturer: Cube Steak Machine Co., Boston.



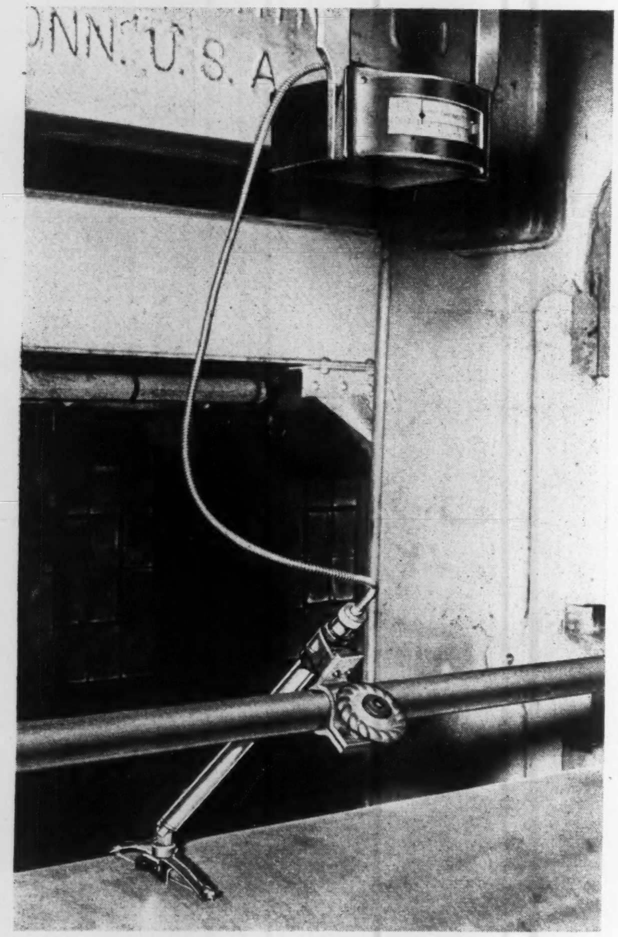
Mechanical Physiotherapist

Power unit of this equipment contains motor and bellows to displace air in shield unit. Aluminum case is flock-lined for sound proofing. Shield is two-section plastic with rubber fittings for neck, arms and lower body to insure air tightness. Manufacturer: Douglas Sales Co., Los Angeles.

(New machines listed on Page 196)

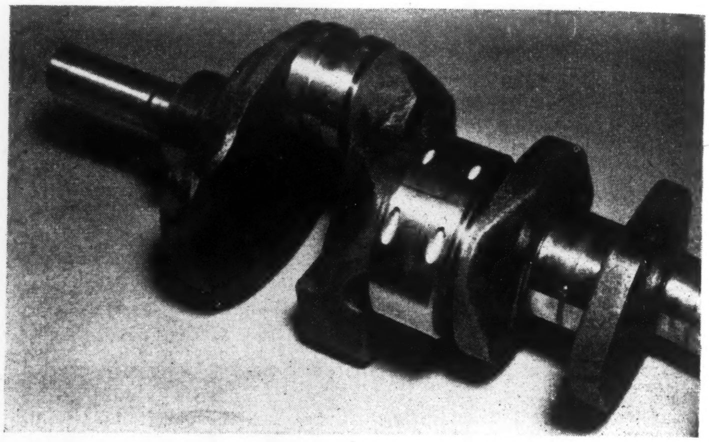
Applications

of Engineering Parts, Materials and Processes



Sliding Thermocouple

ACCURATELY MEASURING the surface temperature of a calender roll, above, the Cambridge roll pyrometer is rigidly attached to a fixed part of the machine. Calendering of materials such as linoleum requires that the temperature of the sheeting calender be held within close limits. A strip thermocouple having the hot junction at its midpoint is held in sliding contact with the roll, providing for a continuous temperature reading at the remotely located indicator here seen above the roll.

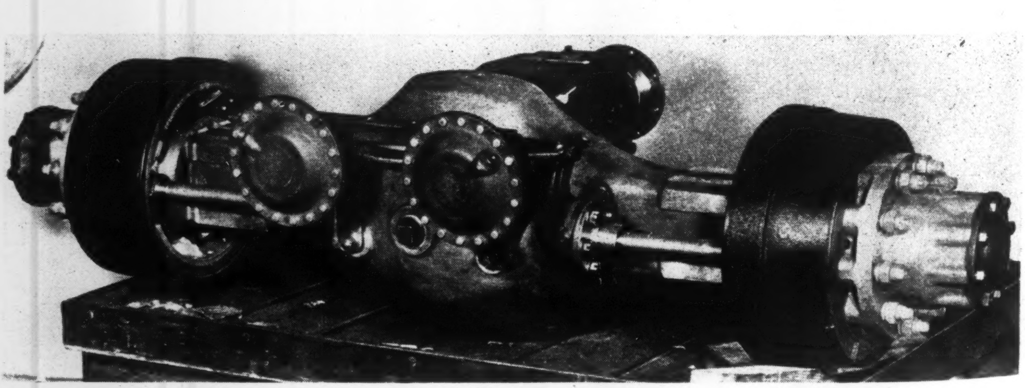


Compact Design

UTILIZING the inherent high-load capacity of ball bearings, in positions hitherto reserved for bulkier journal bearings, the crankshaft design shown above has made possible a lighter, smaller engine. The ball bearings used, manufactured by the Split Ballbearing Corp., are made in halves, machined and ground as assembled sub-units. Consisting of six parts, exclusive of balls and fasteners, the bearings are assembled upon the shaft with eight socket-head cap screws.

Aluminum Axle

WEIGHT SAVING of over 200 pounds is achieved in the heavy-duty truck axle, below, made with aluminum housing, hubs and brake shoes. Designed for axle loads in excess of 18,000 pounds, the unit was developed by the Timken-Detroit Axle Co. to provide improved riding qualities and lengthened tire life. While the production cost is increased because of the alloy used, increase in vehicle pay loads more than compensate for the larger initial investment. Tests have indicated a service life equal to the conventional type of steel axle assembly.



Deflections and Critical Speeds of Stepped Shafts

By H. L. Blood
The Heald Machine Co.
Worcester, Mass.

FAMILIAR formulas for deflections do not apply to beams or shafts of varying section. For such cases different and less familiar methods are frequently used, requiring extra time and study (1) (2)*. While these methods give complete data, sufficient information for many purposes may be obtained by extending the simple formulas, as shown in this Data Sheet, so that the calculations can be made quickly by slide rule.

Formulas for the deflections of stepped beams, such as shown in *Fig.* 1, may be derived quite easily. Starting with the deflection of a beam of uniform section, simply add the increase in deflection which occurs when the moment of inertia of a portion of the beam is reduced. For example, a uniform cantilever with load concentrated at the end, has a maximum deflection of $Pl^3/3EI$. If a portion of the beam is reduced, as in *Fig.* 1*a*, the increase in deflection is

$$\frac{P}{3E}\left(\frac{a^3}{I_1}-\frac{a^3}{I}\right)$$

The sum of these deflections is given in the right-hand column.

These formulas may be extended to cover any number of steps, several other examples being given in *Fig.* 1. The same formulas apply when the moment of inertia is increased, *Fig.* 1*c*. Maximum slope of the deflection curve may be obtained in a similar manner.

Expressions in *Fig.* 1 may be employed to calculate spring constants for the solution of vibration problems, the spring constant being the load divided by the deflection (3).

A beam of the type shown in *Fig.* 1*e* was calculated by Hetenyi (1) by two methods, one approximate and one exact. Using the same values ($W=13.5t$, $l=45$ ft, $a=15$ ft, $I=1.5\,I_1$) in formula *Fig.* 1*e*, the maximum deflection, in ton-foot^3 units, is $12577/EI_1$ which agrees with Hetenyi's exact solution.

MAXWELL'S LAW: Calculations may often be simplified by making use of Maxwell's Law, which says that the deflection of a beam at any point *A* from a load at *B* is the same as the deflection at *B* from the load at *A*. An application of this principle is shown in *Fig.* 1*f*.

DEFLECTION OF TWO-BEARING SPINDLE: *Fig.* 2 gives formulas for calculating the deflection of a two-bearing spindle, with an overhung load *P*. As is customary, the deflection is figured at the point of application of the load, as if no deflection occurred between the end of the spindle and the load. The spindle is considered horizontal at the front bearing B_1. Deflection from the rear bearing reaction B_2 is converted into deflection at the load by the term f/l. Deflection of the portion of the spindle to the right of B_1 is also given at the load. These two deflections are added, units of 0.0001 being employed to simplify calculation.

CRITICAL SPEEDS OF SHAFTS: Since the deflection of a shaft determines the critical speed, the formulas which have been given offer a quick method for calculating critical speeds by slide rule. Sections of a shaft carrying distributed weights are treated in the usual way, that is, the weight at each section is considered as concentrated at the center of gravity of the section.

The shaft shown in *Fig.* 3 supports weights *Q* and *R* which include the weight of the shaft. The formulas may be arranged thus, being deflections upward from point *O*, *Fig.* 3*a*:

$$\text{At } P:\ \frac{1}{3E}\left[P\left(\frac{l^3-a^3}{I}+\frac{a^3}{I_1}\right)-\frac{Qc^2(2l+b)}{2I}\right]=0.000785$$

$$\text{At } Q:\ \frac{1}{3EI}\left[\frac{Pc^2(2l+b)}{2}-Qc^3\right]=0.000063$$

$$\text{At } R:\ \frac{1}{3EI_2}\left[\frac{Sf^2(2n+h)}{2}-Rf^3\right]=0.000195$$

$$\text{At } S:\ \frac{1}{3EI_2}\left[Sn^3-\frac{Rf^2(2n+h)}{2}\right]=0.00159$$

These results are used to make a diagram, as shown in

---

*Numbers in parentheses refer to References at end of Data Sheet.

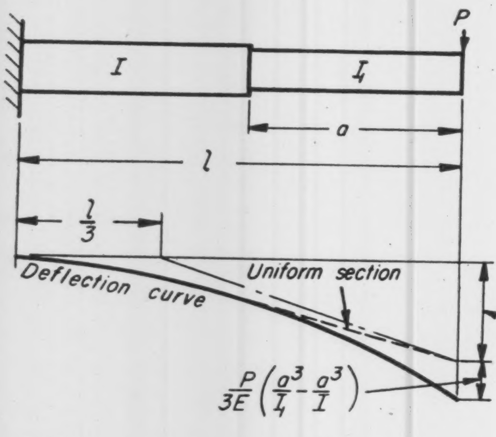
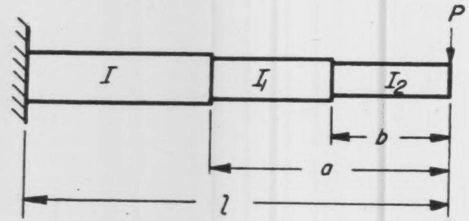
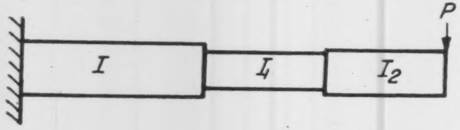
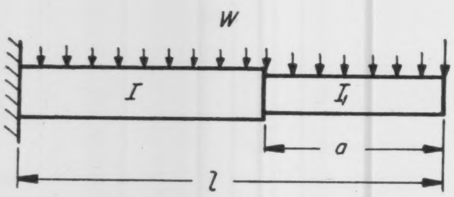
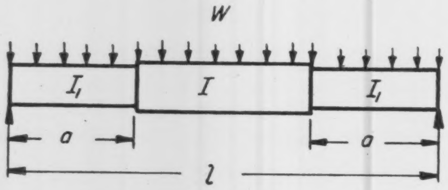
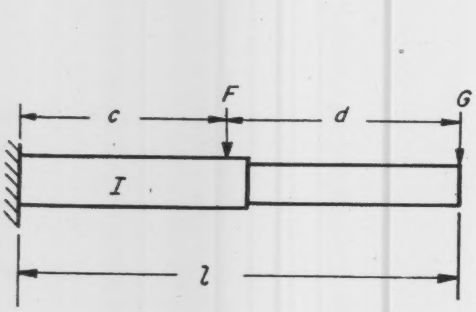
	MAXIMUM DEFLECTION	**MAXIMUM SLOPE**
 P I I_1 a l $\frac{l}{3}$ Deflection curve Uniform section $\frac{Pl^3}{3EI}$ $\frac{P}{3E}\left(\frac{a^3}{I_1}-\frac{a^3}{I}\right)$ (a) I_1 may be greater or less than I	$\frac{P}{3E}\left(\frac{l^3-a^3}{I}+\frac{a^3}{I_1}\right)$	$\frac{P}{2E}\left(\frac{l^2-a^2}{I}+\frac{a^2}{I_1}\right)$
 (b)  (c)	$\frac{P}{3E}\left(\frac{l^3-a^3}{I}+\frac{a^3-b^3}{I_1}+\frac{b^3}{I_2}\right)$	$\frac{P}{2E}\left(\frac{l^2-a^2}{I}+\frac{a^2-b^2}{I_1}+\frac{b^2}{I}\right)$
 (d)	$\frac{W}{8E}\left[\frac{l^3}{I}+\frac{a}{l}\left(\frac{a^3}{I_1}-\frac{a^3}{I}\right)\right]$	
 (e)	$\frac{W}{E}\left[\frac{5l^3}{384I}+\left(\frac{1}{6}-\frac{a}{8l}\right)\left(\frac{a^3}{I_1}-\frac{a^3}{I}\right)\right]$	
 (f)	**DEFLECTION at F from load at G, or at G from load at F** $=\frac{Pc^2(2l+d)}{6EI}$ **regardless of section between F and G.** **Deflections at any point may be added**	**From load at F, slope from F to G** $=\frac{Pc^2}{2EI}$ **From load at G slope at F** $=\frac{Pc}{2EI}(l+d)$

Fig. 3*b*, for the purpose of establishing a base line from which the downward deflections at points *Q* and *R* are measured. Or, the deflections at *Q* and *R* may be obtained by slide rule, as will be apparent from the diagram. These deflections are 0.00097 and 0.0011-in. respectively, which agree closely with those found graphically by Freberg and Kemler(4). Knowing the static deflections *y* at the weights *W*, the lowest critical speed is, according to Timoshenko(5),

$$188\sqrt{\frac{W_1y_1+W_2y_2+W_3y_3}{W_1y_1^2+W_2y_2^2+W_3y_3^2}}$$

$$=188\sqrt{\frac{300\times0.00097+130\times0.0011}{300\times0.00097^2+130\times0.0011^2}}$$

$$=5896 \text{ rpm.}$$

It will be apparent that if the shaft had more steps at the left end, more terms would be added to the formula for deflection at *P*. The difference would be the same as between Case *a* and Case *b, Fig.* 1.

If the shaft had no step at the left end, the first expression would become similar to the fourth. If a step were added at, or to the right of point *R*, the fourth expression would become similar to the first.

An additional weight at *O* would not change the formulas.

Fig. 1—Opposite Page—Typical examples of stepped shafts, with formulas for maximum deflection and slope

Fig. 2—Below—Total deflection of two-bearing spindle is sum of the two for which formulas are given. Deflections are in units of 0.0001-inch, and steel with E=30,000,000 psi is assumed

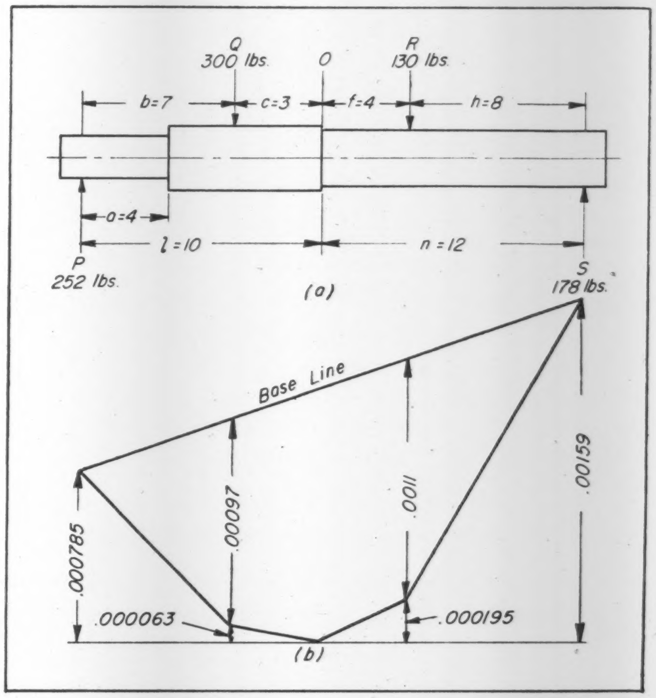
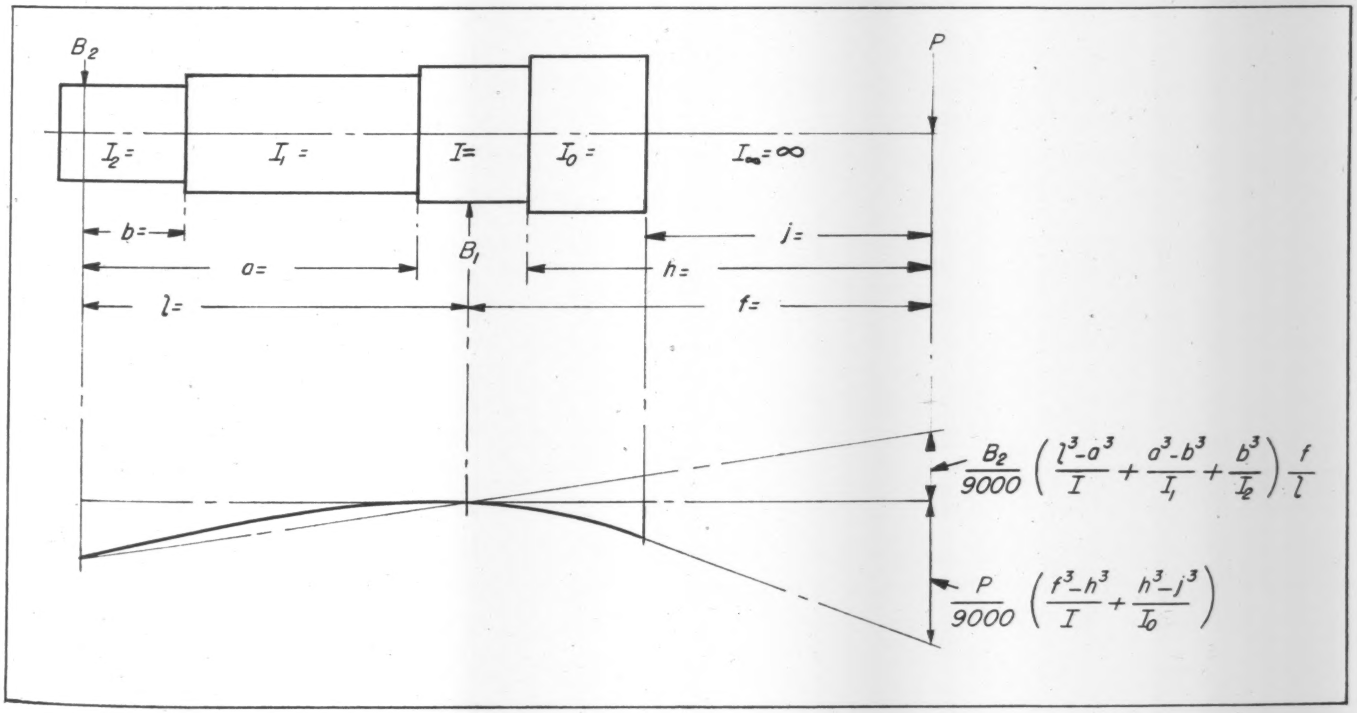

Q
300 lbs
O
R
130 lbs
b=7
c=3
f=4
h=8
a=4
l=10
n=12
P
252 lbs
S
178 lbs
(a)
Base Line
.000785
.00097
.0011
.00159
.000063
.000195
(b)

Fig. 3—Application of deflection formulas for the determination of critical speeds of a rotating shaft

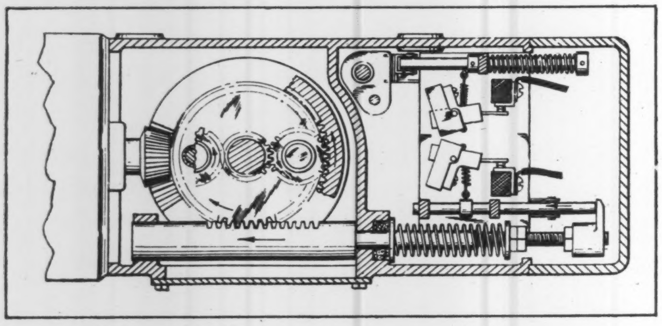
REFERENCES

1. "Deflection of Beams of Varying Cross Section"—Miklós Hetényi, *Trans. ASME,* Vol. 59, 1937, Pages A-49-51.
2. "Stepped Diameter Shafts"—A. F. Puchstein, *Product Engineering,* Dec., 1941, Pages 652-656.
3. "Vibration Problems, II"—J. Ormondroyd, *Trans. ASME,* Vol. 61, 1939, Page A-128.
4. *Elements of Mechanical Vibration*—C. R. Freberg and E. N. Kemler, John Wiley & Sons Inc., New York, Page 77.
5. *Vibration Problems in Engineering*—S. Timoshenko, D. Van Nostrand Co. Inc., New York, Page 95.
6. *Elastic Energy Theory*—J. A. Van Den Broek, John Wiley & Sons Inc., New York, Page 191.


B₂
P
I₂ =
I₁ =
I =
I₀ =
I∞ = ∞
b =
a =
l =
B₁
j =
h =
f =
$\frac{B_2}{9000}\left(\frac{l^3-a^3}{I}+\frac{a^3-b^3}{I_1}+\frac{b^3}{I_2}\right)\frac{f}{l}$
$\frac{P}{9000}\left(\frac{f^3-h^3}{I}+\frac{h^3-j^3}{I_0}\right)$

Noteworthy Patents

OPERATION BEYOND THE POINT where undue torque strain would result is prevented by a torque control unit covered by patent 2,361,439. Assigned to the Vaughan Motor Co. Inc., by Samuel Weiss, the control is particularly adaptable to gate type power valves where excessive strain on the mechanism is undesirable yet a considerably greater amount of power is necessary in unseating than in closing.

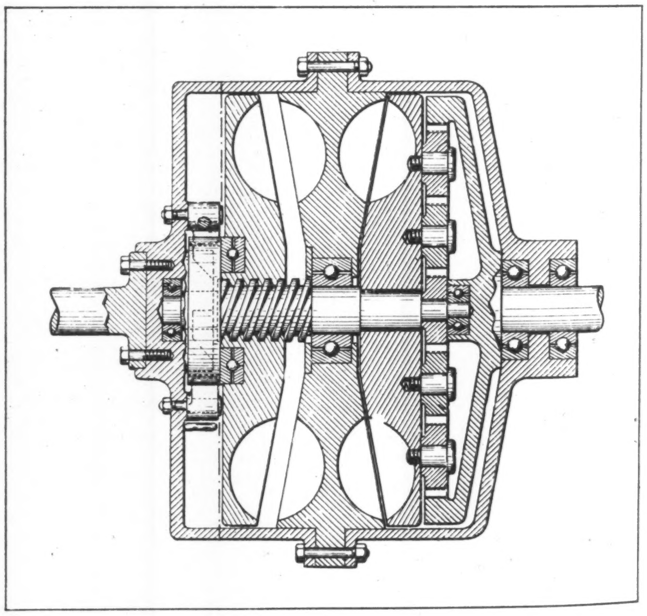


MAXIMUM BRAKING ACTION as well as equalized wear on all brake blocks is assured by a hydraulic brake design outlined in patent 2,390,311 and assigned to Glenn L. Martin Co. by H. P. Kupiel. Sensitive and positive in action, the effect of the brake or "feel" is immediately noticeable on depressing the brake pedal. In addition, the mechanism is said to require less volume of hydraulic actuating fluid than any other equivalent hydraulic braking arrangement.

ELIMINATION OF BACKLASH in a planetary gear train is achieved by an unusual design outlined in patent 2,382,846. Assigned to Bell Telephone Laboratories Inc. by Charles C. Barber, the arrangement provides a gear train wherein backlash between planetary pinions and their associated sun gears is set as desired by adjustment of center distances. Improved planetary pinion support in conjunction with complete adjustment makes possible the drive accuracy necessary with gunfire-control mechanisms, etc., without the objectionable chatter and vibration present with other methods.

TRANSMISSION OF POWER is possible from one end only of a coupling covered in patent 2,371,442 assigned to Gemmer Mfg. Co., by Charles F. Hammond. A machine can be operated freely in either direction of rotation from the drive end of the coupling, but undesirable actuation of the prime mover by the driven member results in instantaneous locking of the driven member of the coupling mechanism.

AUTOMATIC ADJUSTMENT to variations in load is provided by a hydromechanical power transmission mechanism covered by patent 2,385,059. Assigned to the Star Engineering Co. by Henry Buthe, the transmission adjusts itself to load changes through an automatic shift from low-gear drive, effecting maximum application of torque, to high-gear drive, effecting maximum speed commensurate with the load, and vice versa. Transmission of power is flexible, smooth, and free from shock. Drag on the power shaft under idling conditions is reduced to a low value and power shaft acceleration is free from load under starting and pickup conditions until a predetermined rate of speed has been attained, preset according to requirements.



MACHINE DESIGN'S

MATERIALS WORK SHEET

FILING NUMBER
17.02

Plastics Molding Materials

TYPES AND FORMS

CELLULOSE ACETATES AND CELLULOSE ACETATE BUTYRATES: Press polished sheets, films, extruded rods and tubes, special forms, molded parts and extruded shapes.

ETHYL CELLULOSE: Molded parts, sheets and films, rods, tubes, extruded shapes, and hot-melt compositions.

METHYL METHACRYLATES: Molded parts, castings, rods and tubes, sheets, and extruded shapes.

POLYAMIDES (NYLON): Molded parts, yarn, woven cloth, filaments and tubing.

POLYSTYRENES: Molded parts, sheets, rods, tubes, and films.

VINYLIDENE CHLORIDES: Films, molded parts, and extruded rods, tubes and filaments.

VINYL CHLORIDE ACETATES: Molded parts, sheets, tubing, rods, extruded shapes, and films.

MELAMINE FORMALDEHYDES: Laminated sheets, molded parts and adhesives.

PHENOL FORMALDEHYDES: Molded parts, castings, laminated sheets, rods, tubes, varnishes and adhesives.

UREA FORMALDEHYDES: Molded parts, laminated sheets and adhesives.

The plastics molding materials covered in this Work Sheet are those classified by The Society of the Plastics Industry, Inc. in its publication "Classification of Plastics Molding Materials". The society's classification table, reproduced on the following two pages, forms the basis of this Work Sheet. Types of plastics covered are those in general use today.

HOW TO USE THE CLASSIFICATION TABLE

In using the table it is important to note that the values therein listed were obtained at room temperature under standard test conditions. Properties may differ appreciably above or below room temperature. For some plastics and some properties this difference may be negligible, although in most cases it will be appreciable and must be carefully considered if the application involves wide temperature ranges. Also, it should be noted that most tests were conducted in accordance with standard ASTM (American Society for Testing Materials) procedures. ASTM numbers are listed at the top of each column and full particulars regarding test methods may be obtained from ASTM standards.

GRADE NUMBERS: First two digits following the prefix letters SPI indicate the minimum heat distortion temperature using a flexural load of 264 psi fiber stress. For example: 10 for 100 F., 11 for 110 F., etc. The next two digits indicate the minimum Izod impact strength (ft-lb of energy absorbed per inch of notch). For example: 03 for 0.3 ft-lb, 10 for 1.0 ft-lb, etc. The last digit indicates the minimum tensile strength. For example: 5 for 5000 psi, 6 for 6000 psi, etc. Thus, grade number SPI 10232 would express 100 F heat distortion temperature, 2.3 ft-lb impact strength, and 2000 psi tensile strength.

SUFFIX LETTERS: Suffix letters are for use in specifying other properties additional to those controlled by the grade numbers. Capital letters M, E, O, T, C, and A designate mechanical, electrical, optical, thermal, chemical, and aging requirements, respectively. The small suffix letter following the capital letter indicates the specific value desired. For example: Ea for power factor at 60 cycles, Ed for dielectric constant at 60 cycles, etc. Thus, SPI 10232-Eg means that the plastic shall have a minimum heat distortion temperature of 100 F, minimum impact strength of 2.3 ft-lb, and minimum tensile strength of 2000 psi as basic requirements *plus* the added requirement of a minimum 250 volts per mil of dielectric strength, short-time test method. If all of the electrical properties shown as suffix letters Ea, Eb, Ec, Ed, Ee, Ef, Eg, Eh, Ei, Ej and Ek are desired, the capital letter E is used by itself after the grade number. The same procedure applies to the other suffixes such as the capital letters M, E, O, T, C and A denoting all the mechanical, electrical, optical, thermal, chemical or aging properties. Finally, if all the unique properties across the table after any particular SPI grade number are desired, the letter Z is used in the following manner: SPI 10232-Z.

The letter X after each respective grade number denotes the basic chemical composition. This suffix has been included for those who wish to specify the basic chemical composition along with its corresponding grade number. For example, SPI 10232-X means that the material shall have a minimum heat distortion temperature of 100 F, impact strength of 2.3 ft-lb, 2000 psi tensile strength, and shall be formulated from cellulose-acetate-butyrate flake.

CLASSIFICATION OF PLASTIC MOLDING MATERIALS

GRADE NUMBER	BASIC REQUIREMENTS					REQUIREMENTS ADDED BY SUFFIX LETTERS																												GRADE NUMBER
							SOURCE OF SPECIFICATION			MECHANICAL PROPERTIES					ELECTRICAL PROPERTIES											OPTICAL PROP.			THERMAL PROP.		CHEM. PROP.	AGING PROP.		
															Power Factor			Dielectric Constant			Dielectric Strength Volts per Mil			Resistance										
	Heat Distortion Temp. °F 264 PSI Fiber Stress	Jmpact Strength Izod Ft.-lbs. per in. notch	Tensile Strength PSI	Maximum Continuous Service Temp. °F	Heat Distortion Temp. °F 66 PSI Fiber Stress	Material Basic Chemical Analysis	American Society for Testing Materials	Federal	Joint Army-Navy (JAN)	Flexural Strength PSI Fed. Spec. 1031	Compressive Strength PSI	Modulus of Elasticity in Tension PSIx10^6	Hardness Rockwell	Specific Gravity									Volume Resistivity Meg-Ohms cm	Insulation Meg-Ohms	Arc Sec.	Index of Refraction	Light Trans. %	Haze %	Coefficient of Thermal Expansion Linear in/in/°C x10^{-5}	Rate of Burning	Water Absorption % Net Gain	Weight Loss %	Dimension Change %	
	ASTM D648-41T	ASTM D256-43T	ASTM D638-42T (3) D638-44T (3) D651-44T (4)	(2)	ASTM D648-44T	(1)				ASTM D650-42T ASTM D48-43T (5)	ASTM D695-42T ASTM D649-42T (6)	(7)	(8)	ASTM D792-44T (10)	ASTM D150-42T			ASTM D150-42T			ASTM	D149-44T	ASTM D257-38	ASTM D257-38	ASTM D495-42T	ASTM D542-43T	ASTM D672-43T	ASTM D672-45T	(9)	ASTM D635	ASTM D570-42T	(11)	(12)	
															Cycles			Cycles																
															60	10^3	10^6	60	10^3	10^6	Short Time	Step by Step												
	MIN.	MIN.	MIN.	MAX.	MIN.					MIN.	MIN.	MIN.	MIN.	MAX.	MAX.	MAX.	MAX.	MAX.	MAX.	MAX.	MIN.	MIN.	MIN.	MIN.	MIN.	MIN.	MIN.	MAX.	MAX.	MAX.	MAX.	MAX.	MAX.	
						X	Sa	Sb	Sc	Ma	Mb	Mc	Md	Mf	Ea	Eb	Ec	Ed	Ee	Ef	Eg	Eh	Ei	Ej	Ek	Oa	Ob	Oc	Ta	Tb	Ca	Aa	Ab	
SPI 10232	100	2.3	2000		140	CAB	D707-7							1.19							250					1.46	80				1.60	2.4		SPI 10232
SPI 10252	100	2.5	2000		130	CAB	D707-10			3200				1.21							250										1.30	6.0		SPI 10252
SPI 10302	100	3.0	2000		120	CA	D706-11							1.32							250										4.50	11.3		SPI 10302
SPI 10311	100	3.1	1000		130	CAB	D707-8			1600				1.18							250					1.46	80				1.50	3.0		SPI 10311
SPI 11172	110	1.7	2000		140	CA	D706-4			4000				1.37							250										2.90	7.5		SPI 11172
SPI 11173	110	1.7	3000		140	CAB	D707-9			4200				1.22							250										1.30	4.5		SPI 11173
SPI 11182	110	1.8	2000		150	CAB	D707-2			3700				1.20							250										1.70	2.0		SPI 11182
SPI 11203	110	2.0	3000		150	CA	D706-15			3500				1.37							250										1.90	7.0		SPI 11203
SPI 11223	110	2.2	3000		130	CA	D706-10							1.32							250										4.30	9.3		SPI 11223
SPI 11304	110	3.0	4000			EC	D787-4			5000				1.19	0.01	0.03	0.03	4.0	3.8	3.7	350										1.60	2.0		SPI 11304
SPI 11504	110	5.0	4000			EC	D787-7							1.09							250										1.70	1.4		SPI 11504
																										1.46	80							
SPI 12027	120	0.2	7000			VCA	D728-1			12000	9000			1.35									10^8							(13)	0.15			SPI 12027
SPI 12034	120	0.3	4000			VCA	D728-2			7500	9000			1.40									10^8							(13)	0.15			SPI 12034
SPI 12104	120	1.0	4000		160	CAB	D707-3			6100				1.23							250										1.50	1.9		SPI 12104
SPI 12123	120	1.2	3000		150	CAB	D707-4			5000				1.22							250										1.40	3.0		SPI 12123
SPI 12133	120	1.3	3000		140	CA	D706-3			5500				1.37							250										3.10	5.5		SPI 12133
SPI 12144	120	1.4	4000		160	CA	D706-14							1.34							250										2.10	4.3		SPI 12144
SPI 12153	120	1.5	3000		160	CAB	D707-1			5100				1.21							250										1.80	1.5		SPI 12153
SPI 12174	120	1.7	4000		150	CA	D706-8							1.32							250										3.00	5.6		SPI 12174
SPI 12193	120	1.9	3000		140	CA	D706-9							1.32							250										3.50	7.1		SPI 12193
SPI 12355	120	3.5	5000			EC	D787-3			6000				1.18	0.01	0.03	0.03	4.0	3.8	3.7	350										1.60	1.40		SPI 12355
SPI 12505	120	5.0	5000			EC	D787-6							1.12							250										1.70	0.80		SPI 12505
SPI 13015	130	0.1	5000		170	MM	D788-3			12000				1.20	0.07	0.06	0.04	4.0	4.0	3.0	400	350	10^8			1.48	90				0.50	NONE		SPI 13015
SPI 13025	130	0.2	5000		170	MM	D788-1			12000				1.20	0.07	0.06	0.04	4.0	4.0	3.0	400	350	10^8			1.48	90				0.50	NONE		SPI 13025
SPI 13035	130	0.3	5000			VCA	D728-3			8000	9000			1.40									10^8							(13)	0.15			SPI 13035
SPI 13104	130	1.0	4000		160	CA	D706-2			7000				1.37							250					1.46	80				2.80	4.5		SPI 13104
SPI 13104-2	130	1.0	4000		180	CAB	D707-6			6900				1.22							250										1.90	0.8		SPI 13104-2
SPI 13355	130	3.5	5000			EC	D787-2							1.15							350										1.70	0.8		SPI 13355
SPI 13505	130	5.0	5000			EC	D787-5							1.14							250										1.70	0.2		SPI 13505
SPI 14085	140	0.8	5000		180	CA	D706-1			8500				1.37							250					1.46	80				3.50	3.0		SPI 14085
SPI 14085-2	140	0.8	5000		170	CA	D706-7			6000				1.37							250					1.46	80				4.40	3.0		SPI 14085-2
SPI 14115	140	1.1	5000		170	CA	D706-13							1.34							250										2.40	2.5		SPI 14115
SPI 14256	140	2.5	6000			EC	D787-1							1.14							350										1.70	0.2		SPI 14256
						VC	D728-1			15000				1.75	0.042	0.075	0.065	5.1	4.2	3.3	350	300	10^8							(13)	0.10	2.0		SPI 15054
SPI 15095	150	0.9	5000			CA	D706-12							1.34							250										2.70	1.6		SPI 15095
SPI 16015	160	0.1	5000		210	MM	D788-4			12000				1.20	0.07	0.06	0.04	4.0	4.0	3.0	400	350	10^8			1.48	90				0.50	NONE		

SPI 14115	140	1.1	5000		170	CA	D706-13							1.34							250													
SPI 14256	140	2.5	6000			EC	D787-1							1.14							350										1.70	0.2		SPI 14256
						VC	D729-1			15000				1.75	0.042	0.075	0.065	5.1	4.2	3.3	350	300	10^8							(13)	0.10	2.0		SPI 15054
SPI 15095	150	0.9	5000			CA	D706-12							1.34							250										2.70	1.6		SPI 15095
SPI 16015	160	0.1	5000		210	MM	D788-4			12000				1.20	0.07	0.06	0.04	4.0	4.0	3.0	400	350	10^8			1.48	90				0.50	NONE		SPI 16015
SPI 16025	160	0.2	5000		210	MM	D788-2			12000				1.20	0.07	0.06	0.04	4.0	4.0	3.0	400	350	10^8			1.48	90				0.50	NONE		SPI 16025
SPI 16056	160	0.5	6000		210	CA	D706-5			9000				1.37							250					1.46	80				4.70	1.1		SPI 16056
SPI 16069	160	0.6	9000		380	PA	D789							1.15							350										1.50			SPI 16069
SPI 17035	170	0.3	5000	150		PS_1	D703-1			8000				1.07	0.0005	0.0005	0.0010	2.75	2.75	2.75	450	400	10^9	10^4	60	1.58	85	5.0			0.10	0.1		SPI 17035
SPI 17035-2	170	0.3	5000	150		PS_1	D703-2			8000				1.07	0.0003	0.0003	0.0005	2.60	2.60	2.60	500	450	10^{11}	10^6	60	1.58	85	5.0			0.10	0.1		SPI 17035-2
SPI 18023	180	0.2	3000			PS_2	D703-3			4000				1.40	0.002	0.003	0.001	2.80	2.80	2.80	400	400	3.5×10^{13}	10^6	5						0.10	0.1		SPI 18023
SPI 23036	230	0.3	6000	250		PF_1	D700-1			11000	25000			1.30							300			100							0.20			SPI 23036
SPI 24035	240	0.3	5000	300		PF_7	D700-7			8000	15000			2.00	0.050	0.030	0.020	6.0	5.5	5.0	350	300	10^6	300							0.15			SPI 24035
SPI 24035-2	240	0.3	5000	250		PF_8	D700-8			8000	15000			2.00	0.025	0.020	0.010	6.0	5.5	5.0	350	300	10^6	300							0.07			SPI 24035-2
SPI 24085	240	0.8	5000	250		PF_4	D700-4			9000	25000			1.45							200	75		100							1.50			SPI 24085
SPI 24176	240	1.7	6000	250		PF_5	D700-5			9000	25000			1.45							200	75		100							1.75			SPI 24176
SPI 24406	240	4.0	6000	250		PF_6	D700-6			9000	15000			1.45							200	50		100							1.75			SPI 24406
SPI 26024	260	0.2	4000	350		PF_{10}	D700-10			7500	15000			2.00							300	200		300							0.25			SPI 26024
SPI 26026	260	0.2	6000	300		PF_2	D700-2			9000	25000			1.45							300	250		100							0.80			SPI 26026
SPI 26036	260	0.3	6000	300		PF_3	D700-3			9000	25000			1.42							300	200		100							1.00			SPI 26036
SPI 26074	260	0.7	4000	400		PF_9	D700-9			9000	15000			2.00							200	150		300							0.25			SPI 26074
SPI 27025	270	0.2	5000	210		MF_2	D704-2			7500	25000			2.20			0.06	8.0	7.3	6.7	300	250			120						0.15			SPI 27025
SPI 27026	270	0.2	6000	170		UF_1	D705-1			10000	24000			1.55							300	250									3.00			SPI 27026
SPI 27026-2	270	0.2	6000	170		UF_2	D705-2			10000	24000			1.55							300	250									3.00			SPI 27026-2
SPI 27026-3	270	0.2	6000	170		UF_3	D705-3			10000	24000			1.55	0.10	0.06	0.04	9.5	9.0	7.5	300	250	10^6		90						3.00			SPI 27026-3
SPI 30026	300	0.2	6000	210		MF_1	D704-1			9000	25000			1.55	0.08	0.05	0.06	9.5	9.0	8.2	300	250			80						1.80			SPI 30026

SPECIAL NOTE:

Values shown in the table were determined in most cases by using ASTM Test Methods. Property values of molded parts may be higher or lower since the design and molding conditions have an important bearing on the final properties.

THERMOPLASTIC

CA	Cellulose Acetate	PS_1	Polystyrene
CAB	Cellulose Acetate Butyrate	PS_2	Polystyrene
EC	Ethyl Cellulose	VC	Vinylidene Chloride
MM	Methyl Methacrylate	VCA	Vinyl Chloride Acetate
PA	Polyamide		

THERMOSETTING

MF_1	Melamine Formaldehyde	PF_7	Phenol Formaldehyde
MF_2	Melamine Formaldehyde	PF_8	Phenol Formaldehyde
PF_1	Phenol Formaldehyde	PF_9	Phenol Formaldehyde
PF_2	Phenol Formaldehyde	PF_{10}	Phenol Formaldehyde
PF_3	Phenol Formaldehyde	UF_1	Urea Formaldehyde
PF_4	Phenol Formaldehyde	UF_2	Urea Formaldehyde
PF_5	Phenol Formaldehyde	UF_3	Urea Formaldehyde
PF_6	Phenol Formaldehyde		

COEFFICIENT OF THERMAL EXPANSION—PLASTICS AND SOME METALS

(30-60 C, per degree C)

Material	Coef. $\times 10^{-6}$
Phenolics:	
General purpose	39
Medium impact	38
Medium impact—CFI-5	33
High impact—CFI-10	29
High impact—CFI-20	22
Medium heat resistance	34
High heat resistance	26
Electrical, general purpose	35
Low loss	19
Arc resistant	49

Material	Coef. $\times 10^{-6}$
Cellulose acetate	800-1600
Cellulose acetate butyrate	1100-1700
Ethyl cellulose	1000-1400
Urea formaldehyde	250-300
Melamine formaldehyde	250-300
Vinylidene chloride	1900
Methyl methacrylate	700-900
Polystyrene	600-800
Polyethylene	1800
Nylon—Type FM-1	1030
Vinyl chloride acetate	690

Material		Coef. $\times 10^{-6}$
Aluminum 24S-T		21.96
Copper	99.9+	17.71
Phosphor Bronze		16.80
Brass	67 Cu, 33 Zn	18.50
Bronze (commercial)	90 Cu, 10Zn	18.80
Steel	99 Fe, 1 C	12.00
Stainless Steel	90-92 Fe + 8 Cr	11.00
Nickel		12.90
Monel	60 Ni, 12 Fe, 11 Cr, 2 Mn	14.00
Silver (standard)	92.5 Ag, 7.5 Cu	18.00
Nickel Silver (18%)		16.50

CHARACTERISTICS

THERMOPLASTIC PLASTICS

Cellulose Acetates: These are available in transparent, translucent and opaque colors, and in mottled and pearl effects. They soften at from 120 to 180 F. They are well adapted to injection or compression molding or extrusion. Subject to attack by strong acids and alkalies, they are soluble in ketones and esters. Molding compounds generally are furnished in granular form, though molding sheets or molding blanks sometimes are used. Numerous grades of molding powders are available ranging from extra hard to extra soft, the latter types molding at lower temperatures and possessing higher elongation and impact qualities.

Cellulose-Acetate-Butyrates: While similar to the cellulose acetates, they absorb only about half as much water on continuous immersion, are more stable at elevated temperatures for the same degree of flow, are soluble in a wider range of solvents, and compatible with more gums, resins and plasticizers. They are thermoplastic, weather resistant, well adapted to injection or compression molding, and well suited to continuous extrusion. Like the cellulose acetates, they are available in transparent, translucent and opaque colors, and in a variety of mottled effects. Competitive to these materials are the high acetyl content cellulose acetates and cellulose propionates.

Ethyl Cellulose: These thermoplastics have exceptional toughness and flexibility at below zero temperatures. Also, they have good electrical properties, good dimensional stability and are resistant to alkalies, weak acids and moisture. They are suitable for injection molding and extruding, are soluble in many organic solvents except straight-chain petroleum hydrocarbons and are compatible with many resins, oils, waxes and plasticizers.

Methyl Methacrylates: Moldings and castings offer good resistance to discoloration on aging and are strongly resistant to weathering. Also, they are resistant to moisture, alkalies, oils, and acids. Parts may be optically clear and colorless, or translucent or opaque in many colors and coloring effects. Methyl methacrylates have edge-lighting ability, good dimensional stability, are odorless and tasteless, have soft surfaces, and are easily fabricated.

Polyamides: Popularly known as nylon, plastics in this group have high strength, great toughness, good electrical properties, good resistance to chemicals and are readily machined. They are capable of higher operating temperatures than other thermoplastics, do not deteriorate with age, and are not attacked by oils, greases, most solvents, alkalies and weak acids. Particularly suitable for molding into extremely thin-walled parts.

Polystyrenes: While parts usually are injection molded or extruded, they can be compression molded also. These plastics have excellent electrical insulation properties, extremely low water absorption, outstanding dimensional stability and exceptional resistance to most acids and alkalies. Polystyrenes are tasteless and odorless and have good color and luster properties. They are readily soluble in numerous organic solvents.

Vinylidene Chlorides: Have high resistance to acids and alkalies and low water absorption. At room temperature they are resistant to all acids and to all common alkalies except concentrated ammonium hydroxide. They are tough, abrasion resistant, and are odorless, tasteless and nontoxic. They have good dimensional stability and freedom from warpage over a wide range of moisture exposure conditions. Materials ranging from soft and flexible to hard and rigid may be obtained. Films offer a high degree of resistance to moisture vapor.

Vinyl Chloride Acetates: These are available in a wide range of colors in material ranging from transparent to completely opaque. Can be extruded and injection or compression molded. Have good resistance to warpage, shrinkage, cracking, and water absorption. They are highly impact resistant, have only a slight tendency to cold flow, are odorless and tasteless, nontoxic, and nonflammable. Flexible elastomers of vinyl chloride acetate are widely used in the textile field and for electrical insulation.

THERMOSETTING PLASTICS

Melamine Formaldehydes: These have good electrical insulating properties, outstanding arc resistance, good heat resistance, low moisture absorption, and good dimensional stability. Parts are hard, have good scratch resistance, good color stability, heat proofness, and resist the action of cleaning agents. Their excellent weathering resistance makes them suitable for outdoor applications. They can be obtained in wide range of colors and coloring effects, and are odorless and tasteless.

Phenol Formaldehydes: Are among the oldest and most widely used of all plastics. They have excellent resistance to solvents and oils. Molded parts with fabric or cord filler have high mechanical strength. With mica filler they have good electrical insulation properties, and with asbestos filler they have high heat resistance and low shrinkage. Cast phenolics are available in many decorative colors.

Urea Formaldehydes: While these are more expensive than the phenolics, they have unlimited fast-color range. Also, they are odorless, tasteless, nonflammable, and have good electrical properties. Some of the ureas have extremely hard surfaces which are highly scratch resistant. They are resistant to common solvents and are considered proof against attack by alcohol, grease, creams and salves. Their resistance to weak acids, however, is only fair and they are markedly attacked by strong acids and alkalies. They are not as heat or water resistant as the melamines.

APPLICATIONS

THERMOPLASTIC PLASTICS

Cellulose Acetates: Used for machine members where moderate strength is required and heat encountered is not excessive. Cellulose acetates are particularly suitable where appearance is a major factor. Typical applications are: Exterior housings of home appliances, radio and instrument knobs and dials, handles, buttons, bezels, escutcheons, switch plates, grilles, airplane windows, automobile steering wheels, horn buttons, telephone bases, fountain pens, slide fasteners, flashlight cases, tool handles, watch crystals, wire insulation, etc.

Cellulose Acetate Butyrates: Used for parts requiring good impact strength and resistance to distortion under heat and humidity or exposure to weather. Typical parts are: Electric clock and portable radio housings, flashlight cases, steering wheels, shower and spray nozzles, instrument panels, knobs, dials, handles, etc.

Ethyl Cellulose: Having exceptional toughness, good electrical properties and good chemical resistance, these plastics find use in applications such as the following: Vacuum cleaner parts, instrument panel assemblies, steering wheels, flashlight cases, automobile hardware, extruded wire insulation, etc.

Methyl Methacrylates: These thermoplastics are used for such parts as aircraft enclosures, safety shields, dials, decorative accessories, gage glasses, ultra-violet windows, reflectors, magnifiers, transparent containers, lenses, automobile trim, dentures, parts in illuminated instruments, etc.

Polyamides: These are nylon and are particularly suitable for parts requiring great toughness, good strength, and resistance to heat and abrasion. While use of nylon in the past has been confined largely to hosiery and for all types of brush bristles, they undoubtedly will be used for molded machine parts more and more in the future. Typical parts for which nylon already has been succesfully employed include: Water-lubricated bearings, extruded tubes, insulation for electric wire, bristles for industrial brushes, carburetor diaphragms (woven and coated nylon), zippers, screening, and cords for rubber tires.

Polystyrenes: Particularly suitable for electrical parts and parts that must have extremely low water absorption. For certain radio and electrical uses, polystyrenes are equal to fused quartz, which they are supplanting. Typical parts are: Instrument panels, high-frequency insulators, battery cases, chemical containers, colored light reflectors, safety glass, etc. Polystyrenes excellent for low-temperature applications.

Vinylidene Chlorides: Particularly adapted to extrusion. Typical parts are: Gaskets, valve seats, ball checks, medicinal probes, flexible tubing and pipe, conveyor belts, filaments for filter fabrics, screening, etc. Representative injection molded parts are: Spray-gun handles, valve seats, acid dippers, filter parts, nozzle tips, molded abrasive wheels, pipe fittings, rollers, guides, spinneret couplings, etc.

Vinyl Chloride Acetates: Typical applications are: Wire insulation, music records, filter cloth, moisture and grease-resistant linings for piping and conveying equipment, watch crystals, drawing instruments, slide rules, radio and refrigerator dials, coatings for sheet metal parts that must be fabricated after coating, etc.

THERMOSETTING PLASTICS

Melamine Formaldehydes: Particularly suited to parts requiring good electrical insulation properties, and heat and stain resistance. Typical parts are: Name plates, translucent instrument panels, circuit breaker parts, ignition system parts, medical instrument handles, etc.

Phenol Formaldehydes: Excellent for parts requiring high strength, dimensional stability and heat resistance. Representative applications are: Pump parts, valve parts, telephone hand sets, gears and cams (from laminated stock), electrical switch parts, business machine cases or housings, bearings, etc. Typical cast forms are: Clock cases, tooling and forming dies, radio cabinets, etc.

Urea Formaldehydes: Particularly suitable for parts which must have outstanding appearance in color and surface finish. Typical applications are: Scale housings, radio cabinets, clock cases, lamp reflectors, electric razor housings, both general and high-frequency type insulators, escutcheons, handles, etc.

FABRICATION

It will be apparent that wherever feasible the complete forming of molded plastics parts should be effected in the mold. Nevertheless, there are occasions where this is not practicable and some form of machining must be employed to finish the part. In addition, there is of course always the necessity of removing flash, gates, sprues, and runners.

MACHINING:

Since all plastics are poor conductors of heat, cutting tools used in machining them heat up relatively fast and care must be exercised not to use excessively high cutting speeds or feeds. Molded thermosetting ureas and phenolics should be machined only when absolutely necessary. They often contain abrasive fillers and, consequently, dull the cutting tools quickly. Best cutting tools for all types of machining are those of exceedingly hard steels or carbide tipped.

DRILLING:

It is advisable to use drills that have been specifically developed for drilling plastics. These are backed out and cleared frequently to free chips, particularly in drilling deep holes. Use of an air jet directed into the hole can increase the production of a drill twenty to thirty times before resharpening is required.

THREADING AND TAPPING:

For small-diameter internal threads, three-fluted high-speed nitrided and chromium-plated taps seem best and tapping can be done dry. On large sizes, water is a better cutting lubricant than oils or kerosene. Specify tap-hole sizes that will result in a 75 per cent depth of thread. This will help prevent tap breakage and give good clean threads. Also specify a moderate countersink on the top of the hole to be tapped. Generally the American National Standard thread form is preferred to others.

FORMING:

Both thermoplastic and thermosetting sheet stock can be formed into a wide variety of shapes in press dies after heating, the thermosetting types requiring considerably more heat than the thermoplastics.

REMOVING FLASH, SPRUES AND RUNNERS:

This often is done with high-speed-spindle carving machines or by shearing dies in punch presses. Also, fins sometimes are removed in tumbling barrels.

CEMENTING:

Where it is desirable to fasten molded units together to form complete parts, cementing often is employed. However, cementing is successful only where parts of the same type plastic are joined. Dissimilar plastics will not bond properly.

OTHER OPERATIONS:

In addition to the operations discussed in the foregoing, plastics can be turned, milled, sawed, sanded, shaved, sheared, ground, buffed and polished. In general, cutting tools for *thermosetting* molded parts work best with less clearance and more rake than tools used for cutting steel or other metals. For machining *thermoplastic* molded parts, tools having zero rake and plenty of clearance seem best.

ENGRAVING AND MARKING:

Three generally used methods are: Mold engraving, branding and machine engraving. In the first, either recessed or raised letters, numerals or designs are engraved in the steel mold cavity, thus being transferred to the molded part surface during molding. The branding method is a combination of burning and molding into the plastic surface. A steel die under carefully controlled heat and pressure will produce clear, sharp, permanent markings in contrasting, plain or metallic colors to an average depth of 0.010-inch. While the cost of machine engraving runs higher than other methods of marking, there are applications in precision work where it is necessary. Recessed lettering, etc., produced either by mold engraving or machine engraving can, of course, be filled with lacquer or paint of any color desired.

ASSETS to a BOOKCASE

The Automotive Chassis

By P. M. Heldt, member of Society of Automotive Engineers; published by P. M. Heldt, Nyack, New York; 576 pages, 5¼ by 8 3/16 inches, clothbound; available through MACHINE DESIGN, *$6.00 postpaid.*

Fourth and last volume of a series on automotive engineering subjects, this book, much as the others, is intended to serve both as a textbook for students and a handbook or reference work for designers. Should interest lie in learning the plan and function of the various parts of an automotive chassis (other than the power plant) or in the engineering theory and materials involved in their design, the information is included in this comprehensive roundup of design data.

As the industry develops from year to year, automotive design is concerned to successively greater extent with refinements, making it difficult for young engineers to become familiar with the origin and historical development of the various parts. Consequently, the background material presented along with shortcomings of early designs should prove of substantial value and interest.

Development of basic mathematical principles along with fundamental engineering data give a working knowledge of the past and present methods and should assist greatly those designers confronted with more or less similar problems in other fields. Including material on commercial, passenger and war vehicles, the nineteen interestingly written chapters cover all the important units such as chassis layouts, propulsive power requirements, frames, axles, steering gears, drives, brakes, springs, wheels, etc.

□ □ □

Industrial Organization & Management

By Lawrence L. Bethel, Yale University, Franklin S. Atwater, production engineer, Fafnir Bearing Co., George H. E. Smith, lawyer and economist, and Harvey A. Stackman, personnel administrator, Scovill Mfg. Co.; published by McGraw-Hill Book Co. Inc., New York; 790 pages, 5¾ by 8¾ inches, clothbound; available through MACHINE DESIGN, *$4.50 postpaid.*

Functions and universal principles of industrial organization and management with which this book is concerned should be of primary interest to design engineers and professional engineers, especially those whose duties bring them into intimate contact with industrial management problems. The authors treat management as a field of specialization within itself rather than merely an adjunct to engineering in their coverage of methods, costs, industrial relations, budgeting, production control, marketing, office management, etc.

Basic fundamentals developed and presented in this book represent the balanced resultant of personal obs vations and studies by men who as a group repres divergent fields of specialization, such as finance, nomics, industrial relations, production engineering, marketing. The principles treated are the sound, t and tested ones widely used in many industries to These are presented with new emphasis in the light the changed circumstances produced and accelerated the intense industrial activity of World War II. New trends, technological developments and principles which show promise of more general adaptation are considered and discussed.

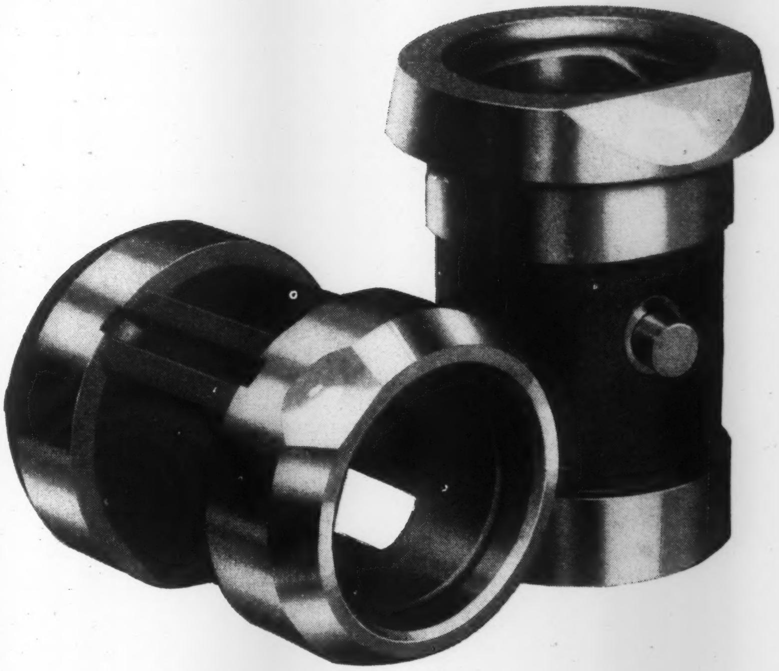
The book is divided into four main sections: I. American Industry—its history and its economic and social foundations; II. Organizing the Industrial Enterprise—the principles of forecasting, financing, internal organization, product development, and physical faculties; III. Operating the Enterprise—manufacturing the product, including the planning of the control functions, selling the product, managing the general offices, and administering the personnel; IV. Coordinating the Enterprise—the dynamics of an industrial organization.

Confronted with the ever-changing and increasingly complex problems in managing industrial enterprise, engineers can ill afford to overlook thorough consideration of these principles of industrial relations, methods analysis and production planning.

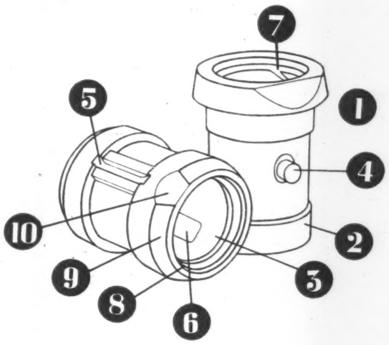
A comprehensive pocket-size volume covering the selection and application of relays or timers has been prepared by the engineering department of Struthers-Dunn, Inc. Based on more than 25 years of highly specialized experience in the solution of relay and timer problems, the handbook is a direct answer to the multitude of queries received concerning their application.

Entitled, "Relay Engineering", the book covers all the fundamental principles involved, equipment and circuits used with relays, servicing, inspection, and a great many every-day as well as specialized relay applications.

Written in clear, simple language, the material is generalized to an extent which makes it applicable to any and all problems regardless of the particular relay design involved. Profusely illustrated with 863 diagrams, 81 tables, and numerous photographs and drawings, it contains 640 pages, 4¼ by 6½ inches, and is available, flexible leatherette bound, from Struthers-Dunn, Inc., 1321 Arch St., Philadelphia 7, for $3.00.



How to tell QUALITY *in* Sleeve Type Bearings



The final proof of QUALITY in Sleeve Bearings is the performance delivered. Performance, however, depends to a very great extent on how the bearings are made. These textile mill bearings are a good example. The first consideration . . . 1 . . . is the alloy. It must contain all the necessary ingredients . . . in their correct proportion . . . to provide the required bearing characteristics. Next, the machining of the outside . 2 . . . and inside . . . 3 . . . diameters must be smooth, and to precise measurements. Lugs, recesses and slots . . . 4-5-6 . . . must be correctly placed, accurately machined to the proper height or depth and free from burrs. Oil grooving . . . 7 . . . must be the proper style, the correct width and depth to guarantee proper distribution of the lubrication. The chamfer . . . 8 . . . must be at the proper angle, as should the self-aligning feature . . . 9 . . . and the flat . . . 10 . . . These are but a few of the many factors that enter into the production of top quality bearings. The easiest way to be sure of getting the finest quality available is to place your bearing requirements with Johnson Bronze. We have the skilled help . . . the complete facilities . . . and more then forty years of bearing experience. Why not consult with us now?

Where to get **QUALITY**

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CLEVELAND
DALLAS
DETROIT
KANSAS CITY

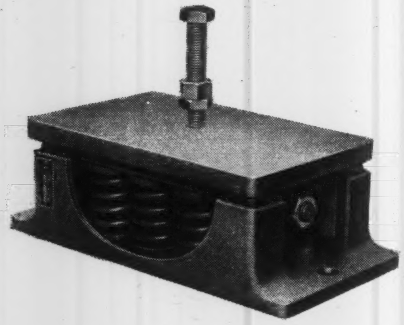
JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS

LOS ANGELES
MINNEAPOLIS
NEWARK
NEW YORK
PHILADELPHIA
PITTSBURGH
SAN FRANCISCO
ST. LOUIS . . . . SEATTLE

NEW PARTS AND MATERIALS

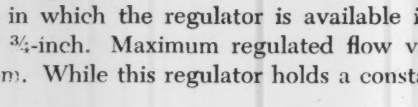
Vibration Mounts

STEEL SPRING TYPE antivibration mounts of a simplified type have been announced by the Korfund Co., 48-15 Thirty-second Place, Long Island City, N. Y. Identified as type LK, the mount is available in six types having ratings from 75 to 12,000 pounds. The unit, which is constructed of either semisteel or welded steel, supports the load upon a platform supported by helical steel compression springs. Longitudinal forces are resisted by resilient compression chocks. Leveling and securing of the load is attained by a bolt protruding from the top of the mount, and bearing upon the springs. This vibration and shock control mount is recommended for presses, grinders, generators, testing and other machines.



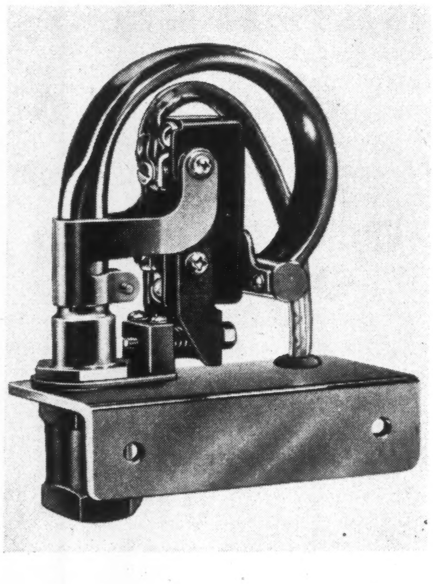
Hydraulic Flow Regulator

BY MEANS OF a calibrated orifice in a floating piston whose movement is resisted by a spring, a new hydraulic flow regulator holds the rate of flow of fluids constant at any desired predetermined amount. The rate of flow will move the piston, throttling escape ports. An excess of flow over the predetermined amount will cause piston to increase throttling, while any momentary decrease in flow will cause the piston to decrease the amount of throttling. Pipe sizes in which the regulator is available include ¼, ⅜, ½ and ¾-inch. Maximum regulated flow varies from 3 to 12 gpm. While this regulator holds a constant rate of flow, the manufacturer, Waterman Engineering Co., 721 Custer Ave., Evanston, Ill., also furnishes special-purpose flow regulators in which the rate of flow varies with the pressure.



Pressure-Actuated Switches

BOURDON-TUBE controlled, a line of "Starbird" pressure-operated switches has been announced by the Meletron Corp., 950 North Highland Ave., Los Angeles 39. Utilizing the pressure-produced deflection of a Bourdon tube to actuate a Micro switch, the device is claimed to be unaffected by tilting or vibration. Being resistant to corrosion, its use is indicated in the food, chemical, and petroleum industries. Stock models are available to operate on pressure from 25 to 3000 psi, actuating single-pole, single-throw; or single-pole, double-throw Micro switches rated to 1380 kva.



Stainless-Steel Electrodes

ELECTRODES for shielded-arc welding of stainless steels, have been announced by the Lincoln Electric Co., Cleveland. Said to be improved, the electrodes known as "Stainweld A 7" and "Stainweld A 7-Cb", are for all-position welding of steels of the 18 per cent chromium and 8 per cent nickel type. "Stainweld A 7", available in six sizes, is recommended for use with steels designated by

SARAN braid

New protection against abrasion,

rot, oil, heat, cold and moisture

Wherever electrical wire and cable demand the utmost in protection—there's a job for Saran (pronounced Sah-*ran*).

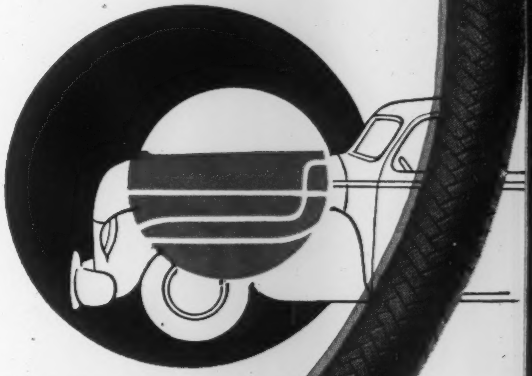
Braid, woven from Saran, a Dow plastic, cuts replacement costs because its toughness resists all forms of damage longer than most cable coverings. It's insurance against rot. It's fungus and mildew proof. It does not absorb moisture and oil—an important quality in such applications as automobile wiring. It's unaffected by a wide range of temperatures. And it resists abrasion far beyond requirements of most jobs.

These characteristics give Saran braid first call for hundreds of applications. Saran braid is new, but it's proving a big advance in wire protection—in simple jobs like telephone and extension cords and in complex radio and radar installations.

Any Dow sales office can tell you how Saran braid will cut down your electrical wiring troubles—and operating costs, too!

Saran braid's flexibility and wearing quality recommend it for telephone cords.

Rough use given such items as extension cords calls especially for Saran braid.



Saran braid protects wiring against oil moisture and temperature extremes.

PRESENT AND POTENTIAL USES: Braided covering for wire and cable of all kinds. Especially valuable in guarding against abrasion in such applications as extension cords, etc. Many uses for cable such as household appliance cord that is exposed to moisture and consequent rendering of shock. Resistant to rot caused by entrapped moisture or oil in such applications as automotive wiring.

PROPERTIES AND ADVANTAGES: Combines toughness with flexibility even at temperature extremes. Flexural strength, p.s.i. 15,000 to 17,000. Highly resistant to broad range of chemicals. Water absorption less than 0.1%. Nonflammable. Excellent electrical insulator. Not subject to distortion or flow at contact points. Available in many colors.

THE DOW CHEMICAL COMPANY • MIDLAND, MICHIGAN

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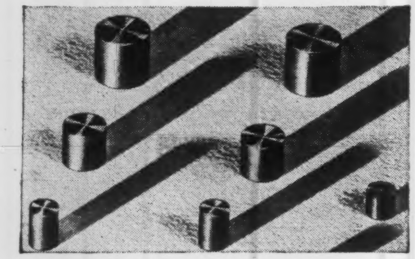
DOW
PLASTICS

STYRON • STYRALOY • ETHOCEL
ETHOCEL SHEETING • SARAN • SARAN FILM

the American Iron and Steel Institute as numbers 304 and 308, and is suited for operation on direct current or higher voltage alternating current. "Stainweld A 7-Cb", is columbium stabilized, and is recommended for use with stabilized 18-8 stainless steels designated as AISI numbers 321 and 347. The electrode is available in four sizes.

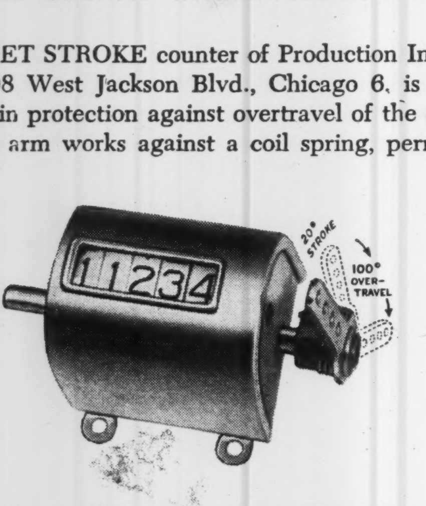
Welding Tips

RESISTANCE WELDING tips, said to possess excellent heat and electrical conductivity, have been announced by the Stackpole Carbon Co., St. Marys, Pa. Available in two standard types, RW-1, and RW-2, the tips were developed to retain hardness under severe working conditions. Both materials can be supplied to practically any required shape or size, and are made to the usual tolerance.



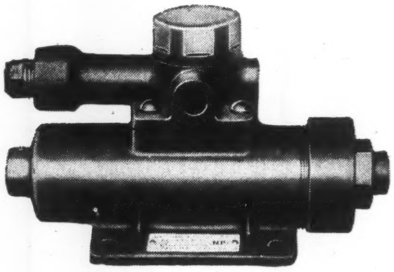
Stroke Counter with Overtravel Arm

NONRESET STROKE counter of Production Instrument Co., 702-08 West Jackson Blvd., Chicago 6, is designed with built-in protection against overtravel of the operating arm. This arm works against a coil spring, permitting it to continue beyond the full stroke without injury to the counter. It may be installed on either side of the counter and set at any angle. Enclosed in a small tamper-proof case, the counter has five large number wheels, registering to 99,999 and repeat.

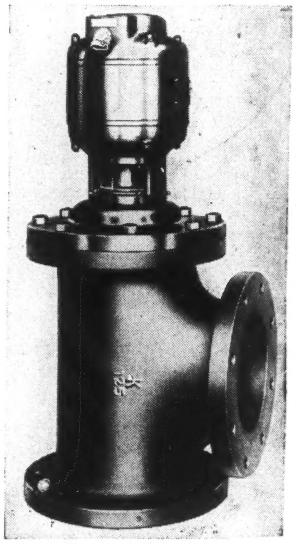


Automatic Water Ejector

RECENTLY DEVELOPED by National Pneumatic Co., 420 Lexington Ave., New York 17, a new automatic water ejector removes water automatically from compressed air systems and does away with frequent manual draining of water separator and air lines, preventing water damage to pneumatic equipment. No loss in pressure is caused inasmuch as the ejector does not connect the compressed air system to atmosphere. Mounted horizontally in any location where air-line water is collected, the ejector is 7 11/16 x 2 3/4 x 4 1/2 inches, and has three connections—of 1/4-inch pipe or 3/8-inch copper tubing—one to bottom of water collector; another to drain; and the third to an air line that is repeatedly charged and discharged. A 1/2-inch x 6 inches pipe and fittings provide a vertical sump of ample capacity for most water removal applications. At 100 pounds pressure, 30 cycles will discharge 1 quart of water. By lengthening the pipe greater quantities can be handled.



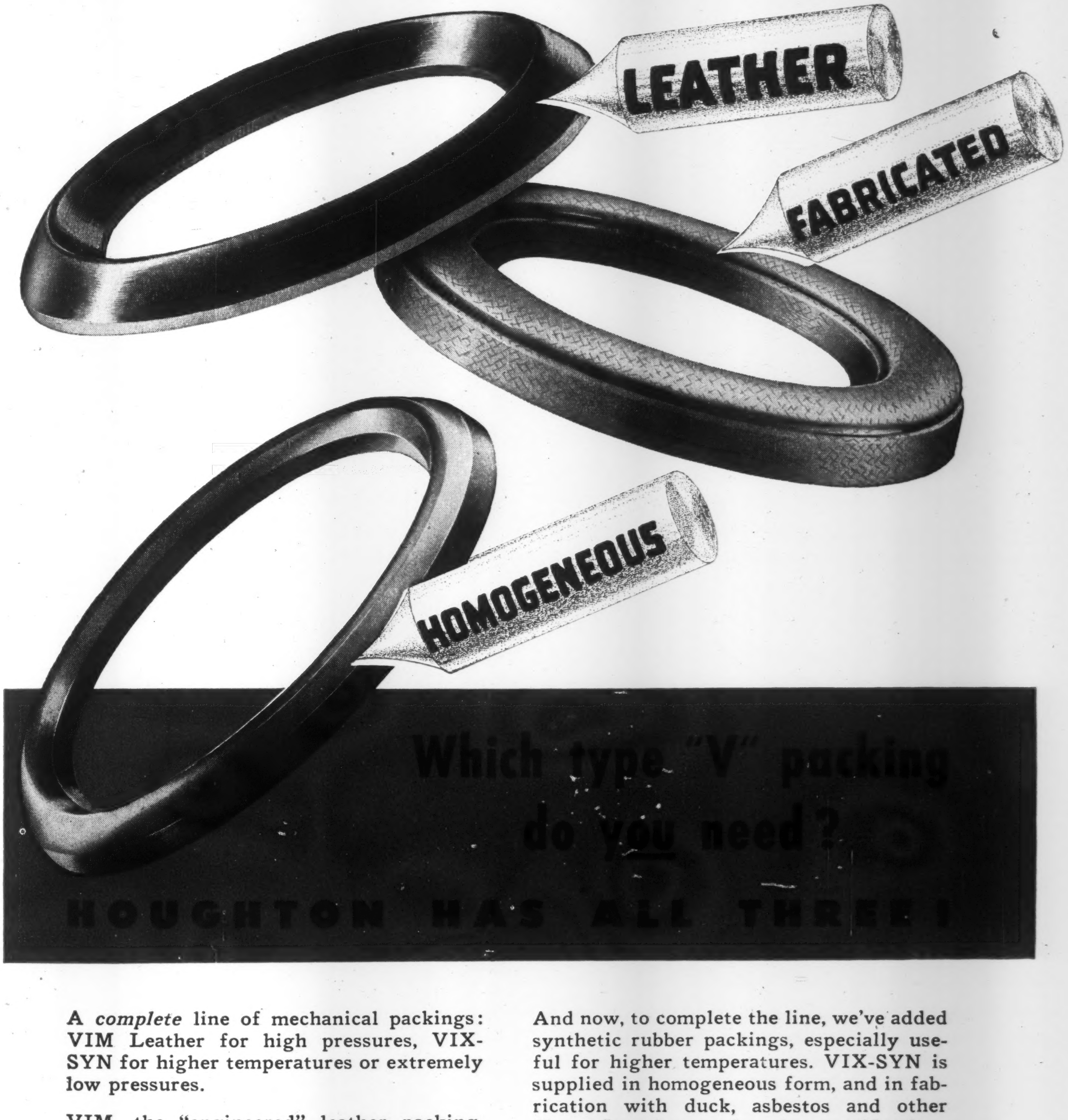
Circulating Pumps



PROPELLER-TYPE low-head circulating pumps are now being manufactured by the Ruthman Machinery Co., 1809 Reading Road, Cincinnati 2, for attachment to standard 125-pound cast-iron pipe tees. Sold under the tradename of "Axiaflo", the circulators are designed for handling large liquid volumes at low head with minimum power requirements. The units are equipped with ball bearing, totally enclosed, dynamically balanced motors. Sizes available are 4-inch with a ¼-hp motor, and 6-inch with ½-hp motor, capacities range from 280 to 1200 gallons per minute in four steps.

Deflection Pick-Up

INTRODUCED to the market by the Stevens-Arnold Co., 22 Elkins St., South Boston, is a hermetically sealed electric deflection unit capable of accurate measurement of force or deflection. Accurate readings are obtained in the range of 0.0005″ to 0.1″ movement of the plunger, and reliable response up to 100 cycles per second is claimed. Because the pick-up output is 75 millivolts, and its



Which type "V" packing do *you* need?

HOUGHTON HAS ALL THREE!

A *complete* line of mechanical packings: VIM Leather for high pressures, VIX-SYN for higher temperatures or extremely low pressures.

VIM, the "engineered" leather packing, has long been the standard for most hydraulic and pneumatic requirements. A war-born development in the VIM line is the resin-impregnated "V," which resists oil so effectively. There are many special impregnations for the long wearing VIM leather, moulded to all standard shapes.

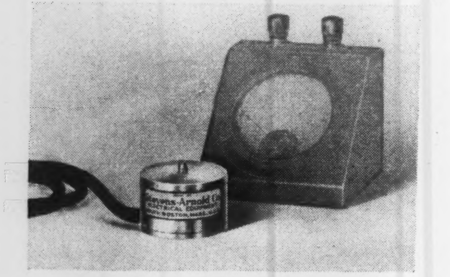
And now, to complete the line, we've added synthetic rubber packings, especially useful for higher temperatures. VIX-SYN is supplied in homogeneous form, and in fabrication with duck, asbestos and other fibres. Complete engineering specifications are available upon request.

So when you're packing-minded, remember HOUGHTON as headquarters for better packings. For that next design need, write E. F. HOUGHTON & CO., 303 W. Lehigh Ave., Philadelphia 33, Penna.

HOUGHTON'S

VIM Leather and VIX-SYN Packings

internal resistance less than one ohm, it may be connected directly to standard indicating instruments of a type that are rugged and portable. Several pick-ups may be connected together in parallel. The pick-up may be used under unfavorable operating conditions including heavy vibration, at temperatures up to 300 F, and because it is hermetically sealed, it may be submerged in water. Power supply may be 115 volts alternating current, or 8, 12, or 24 volts direct current. Results may be recorded on a continuous strip chart.

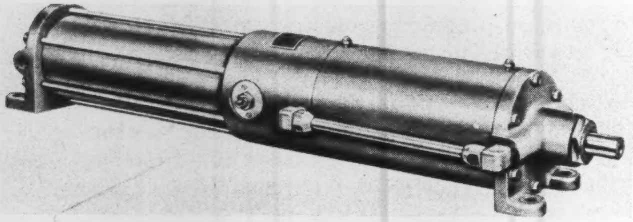


Packaged Arteries

PACKAGED kits of preformed tube and hose sections and fittings have been announced by the J. N. Fauver Co., 49 W. Hancock St., Detroit. Available in tube sizes from ¼-inch to 2½-inch, subassemblies are provided cut and bent to shape. The assembler need only tighten one nut at each end of the fitting. Complete kits are available for hydraulic, coolant, lubrication or other tube systems. Parts may include gages, valves, pumps, meters and filters, with working pressures to 6000 pounds. Items are marked for quick identification.

Pneumatic Cylinders

AIR powered, nonrotating cylinders incorporating hydraulic speed control have been announced by the Logansport Machine Co., Logansport, Ind. The cylinders which are air driven, are said to provide the smooth feed regulation of hydraulic cylinders. An answer is thus said to be provided to the demand for a cylinder in mechanical operations where accurate control of movements is essential but air operation is preferred. The cylinders are made with separate air and hydraulic pistons, assembled as an integral unit on a common piston rod. Rod movements are powered by the air cylinder. Movement of the piston rod forces oil to flow between the chambers of the hydraulic cylinder. An adjustable, built-in valve in this circuit regulates the flow of oil, and hence the speed of the piston rod movement. No hydraulic power unit is used.

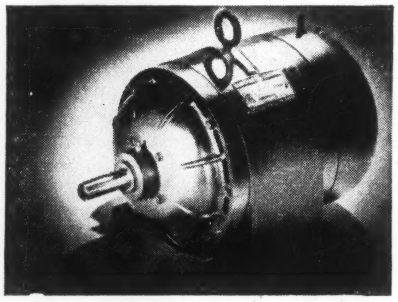


Synthetic Finish

ANNOUNCED by Roxalin Flexible Finishes Inc., 800 Magnolia Ave., Elizabeth, N. J. is a chlorinated-rubber base synthetic finish, said to be particularly applicable for corrosive conditions. Resistant to temperature variations from —40 F to 200 F, the finish is also resistant to moisture, as well as ordinary cleaners, lubricants, or refrigerants. The same formula will air dry or bake and may be applied by spraying or dip tank.

Totally-Enclosed Motor

A NEW totally-enclosed, fan-cooled motor especially designed for use in extremely dusty, dirty, and corrosive atmospheres has been added to the line of General Electric Tri-Clad induction motors. The new motor is available in standard explosion-proof and dust-proof types from 1 to 1000 hp, and can be used where iron dust and metal filings are in the air, and in Class I Groups C and D, Class II Groups E, F, and G conditions. Short in length and compact in construction, the motor can be installed in a small space, making it suitable for machine tool applications where the motor must be part of the driven machine. Low starting current, with balanced design makes the new motor suitable for full-voltage starting, thus permitting the use of simple, inexpensive control equipment. The motor has high pull-up torque, and high maximum running torque to meet temporary abnormal peaks and low voltage conditions.

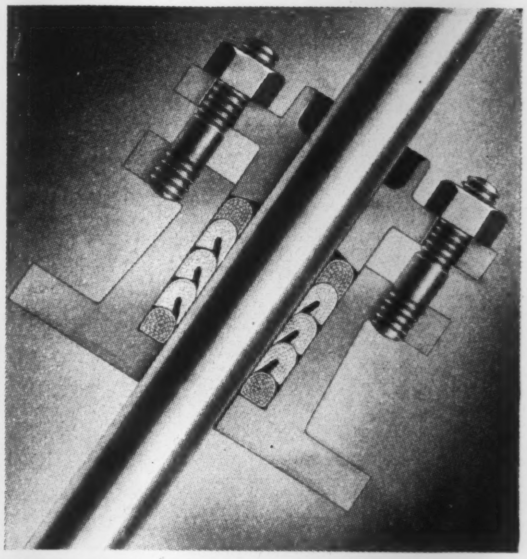


Hydraulic Power Unit

A COMPACT hydraulic pump designed for wide application on tractors has been announced by the George D. Roper Corp., Rockford, Ill. Known as the "Roper Pac", the unit contains three fundamental elements: a precision built rotary gear pump, control valves, and an oil reservoir. Control valves are designed to insure long pump life. An unloading feature prevents damaging heat build up and

Design for

FEWER PACKING REPLACEMENTS



The flexible lips of Sea Rings are forced flat against the rod by pressure on the work stroke. On the return stroke, however, as pressure is removed, the lips automatically release, thus reducing friction and rod wear.

with long-lasting J-M Sea Rings

JM
No. 1 PACKING for RODS and PLUNGERS

ENGINEERING DATA FOR THE PRODUCT DESIGNER

Suggested Typical Flange Widths (Packing Space) of Sea Rings		Depth of Stuffing Box required for 2 Sea Rings, Header & Follower	Added Depth needed for each additional Sea Ring
For Rod Dia.	Flange Width		
3/4" to 1"	5/16"	1 1/2"	3/8"
1" to 3"	3/8"	1 13/16"	7/16"
3" to 5"	1/2"	2 1/4"	17/32"
5" to 7"	5/8"	2 15/16"	3/4"
7" to 10"	3/4"	3 1/2"	15/16"

WHEN YOU KNOW that a set of packings is made to give years of service *without replacement*—you can design machine parts accordingly.

Johns-Manville Sea Rings are uniquely constructed to give long service on reciprocating rods and plungers; they *seal automatically on the work stroke,* yet *release on the return stroke* . . . saving wear on both packing and machine. This reduces costly replacements and shutdown of equipment.

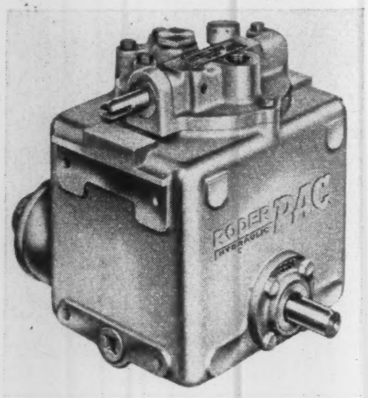
These are reasons why machine designers keep J-M Sea Rings in mind when they plan for economical operation. Custom-made for individual service requirements, temperatures, pressures and fluids, this packing provides a tight seal with an absolute minimum of friction.

Consult the chart on this page . . . or ask Johns-Manville (packings specialists for over 75 years) to help with your design problem. Write Johns-Manville, Box 290, New York 16, N.Y.

JOHNS-MANVILLE JM PRODUCTS

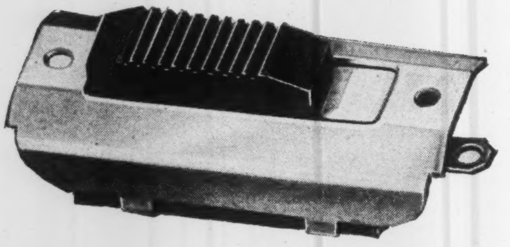
Johns-Manville **PACKINGS & GASKETS**

overloading. When controls are once set in the "raise" position, the unit retains itself in this operating cycle, unless manually changed or overloaded, until the maximum stroke is reached. A valve then automatically unloads the pump. The load is securely held by a check valve, preventing slipping back. When needed, extra power is obtainable for a short duration by holding in reset position. As a safety feature, the operating control always retains itself in a nonoperating cycle until manually actuated.



Snap Switch

BASED UPON a torsional spring design, a snap-action switch identified as the E-4 has been announced by the Allied Control Company, 2 East End Ave., New York 21. Featuring snap action with high contact pressure, the switch is constructed with a laminated bakelite base, and formed steel housing. Models available are single-pole, double-throw, or single-pole, single-throw, in normally-open; and single-pole, single-throw, normally-closed. Measuring 1 5/16-inches long, 5/8-inch high, and 9/16-inch wide the switch is rated at 300 watts alternating current and 86 watts direct current.



Internal Plating of Pipe

INSIDE OF steel pipe can be electroplated with nickel or other metals by means of a new process developed by Bart Mfg. Co., 227 Main St., Belleville 9, N. J. Known as the Lectro-Clad process, it produces a smooth, ductile, pore-free nickel deposit adherent to the base metal and can be applied to pipe or tubing up to 18 inches overall diameter, in lengths approximately 20 feet. Internally plated pipe can be welded, reduced or bent, hot or cold, without destroying the internal lining. In reducing tubing by cold reduction it is possible to predetermine the actual thickness of plating throughout the entire reduction because the applied metal and the base metal reduce in equal ratio. The internally plated pipe will find use in the production or handling of petroleum, chemicals, foods, textiles, plastics, coke and gas, glue, gelatin, sugar, varnishes, paper, glass, ceramics, etc.

Industrial Timer

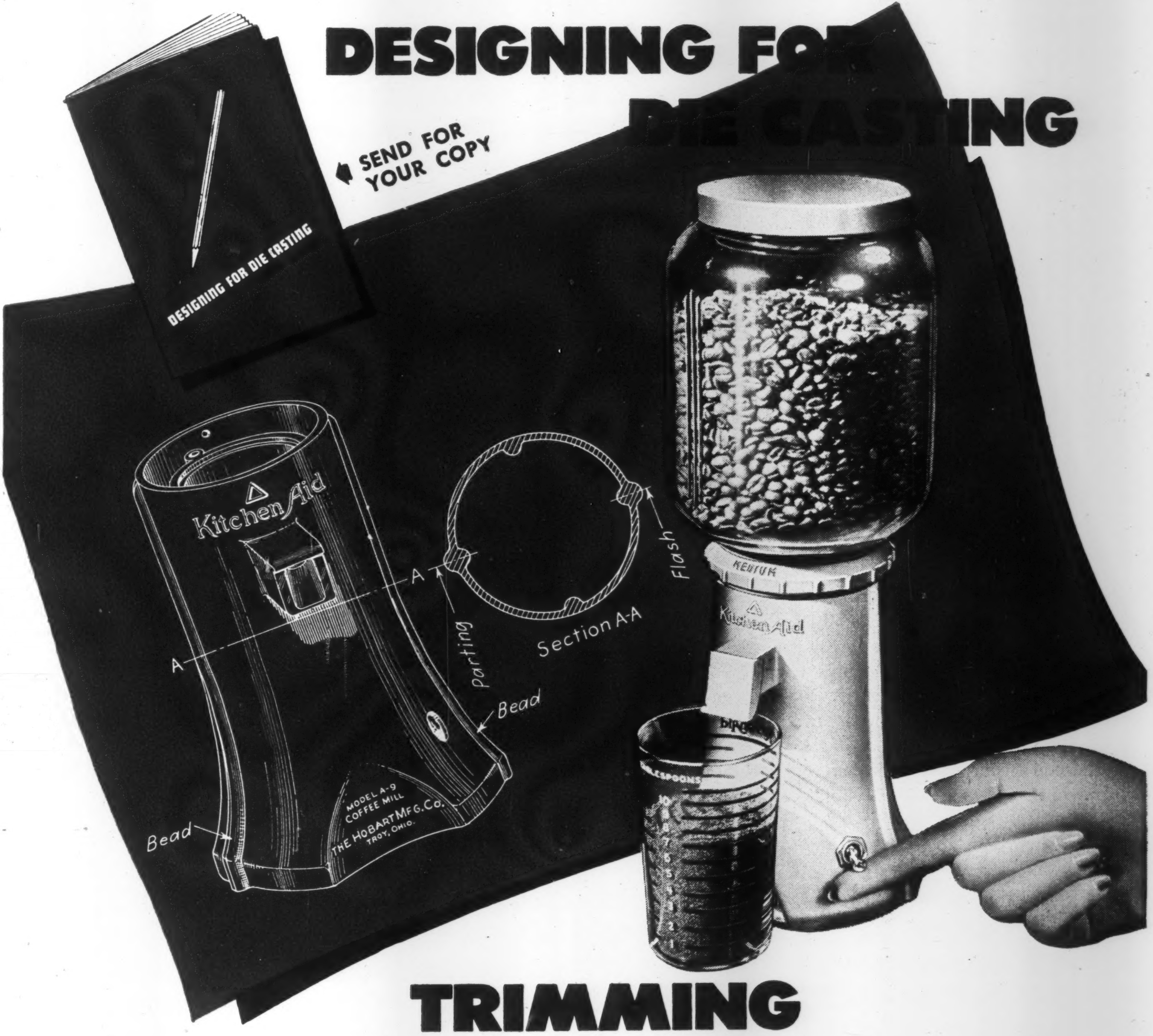
PROVIDING A simple accurate automatic means of controlling the time of exposure on photographic and other apparatus where split-second time gaging is essential, a new model Time-O-Lite instrument has been announced by Industrial Timer Corp., 117 Edison Place, Newark 5, N. J. The new P-49 Model is equipped with an additional receptacle which can be attached to a printer platen switch for setting at the time interval previously selected. Operator can use a foot switch control if the instrument is not adjacent to the enlarger or apparatus. The double scale on the dial has large, clear divisions and numbers. Outer scale is for setting the time interval, with a large white pointer. The inner scale with an added red pointer moves counter-clockwise from the set point to zero, and permits accurate determination of the exact amount of time elapsed.

60
0
5
10
15
20
25
40
45
50
55
TIME
FOCUS
PUSH
TIME-O-LITE
PROFESSIONAL

Silicone Gaskets

POSSESSING heat resistant qualities and flexibility at both high and low temperatures, silicone gaskets manufactured by the General Electric Co. are said to outlast other types of gaskets now commonly used on industrial heating ovens. The resiliency of the material after long exposure to high temperatures will help maintain standard, uniform conditions by preventing entrance of cold air or leakage of hot air in ovens maintained under vacuum or pressure. The softness and resiliency of the silicone rubber product is also said to eliminate the need for close machining of the steel usually required to produce a tight

DESIGNING FOR DIE CASTING



TRIMMING

In designing die castings, it is important to remember that flash always occurs on castings at the die partings, and that steps can be taken in design to bring the flash where it can be trimmed most easily and quickly by a shaving die.

Cost of flash removal is minimized when:

1. *The die parting can be in a single plane.*
2. *Slides and movable cores are not required in the die.*
3. *Cores do not join or intersect in forming the casting.*
4. *Blind holes rather than through holes are employed.*
5. *Contours of the casting at the parting are of simple shape.*
6. *Flash occurs at points where other machining is required, in which case a separate flash removal operation is avoided.*

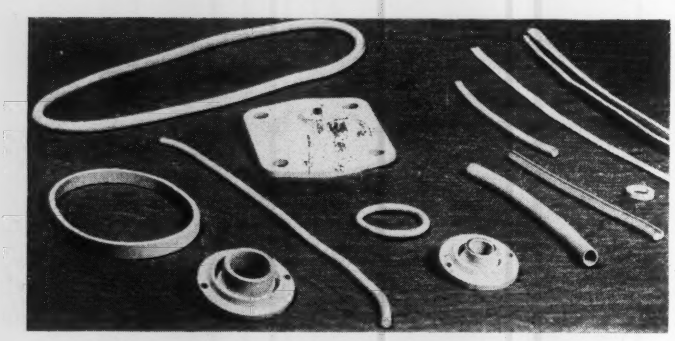
Flash which runs along a flat surface, and is not at the extreme edge of the casting, is difficult to remove cleanly without leaving tool marks on adjacent surfaces. The designers of the above *zinc alloy* die cast coffee mill housing avoided this problem by merely employing decorative beads on either side of the casting at the parting line. Thus the flash occurs on the beads, from which it is easily shaved off without marking the surrounding areas.

Additional data on trimming and other design considerations appear in our booklet "Designing For Die Casting." To insure that you will get the most from your die casting dollar, ask us—or your die casting source—for a free copy of this booklet.

ZINC
FOR DIE CASTING ALLOYS

THE NEW JERSEY ZINC COMPANY • 160 Front St., New York 7, N. Y.

The Research was done, the Alloys were developed, and most Die Castings are based on
HORSE HEAD SPECIAL (99.99+% Uniform Quality) ZINC



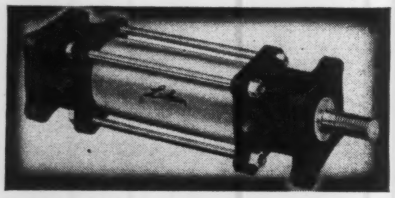
fit between the door and seat. The material, which is available in a variety of extruded shapes and sizes will not adhere to other material in spite of its use under high temperature and oxidizing conditions. Hardness of the material is said to increase only slightly after exposure for one year at 300 F.

Insulating Varnish

A NEW internal-curing insulating varnish, which is claimed to cut curing time up to 50 per cent yet which involves no change in curing equipment, has been announced by Irvington Varnish and Insulator Co., Irvington, N. J. Known as Harvel 912C, it is a new member of the Harvel series of phenol-aldehyde varnishes produced by Irvington. Typical operational curing times of the 912C varnish based on actual coil curing have been two to four hours at 285 F, and four to six hours at 250 F, values varying slightly with coil size and iron mass.

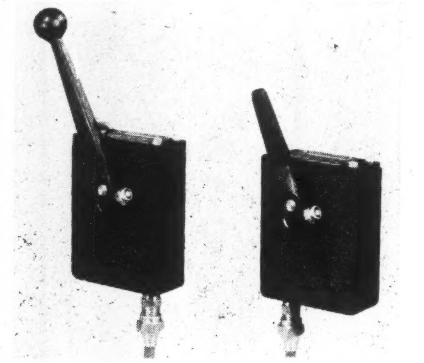
Cylinders

STANDARDIZED cylinders for use with air, oil, or water have been announced by the Engineering Products Co., 1600 S. San Pedro St., Los Angeles 15. Available from stock in 750 possible combinations, the cylinder is made in five diameters and five strokes. Mounting has been simplified by standardizing upon four types of rod attachments and an equal number of head attachments. Having bronze cylinders and semisteel heads, the cylinders are rated to 14,000 pounds pulling capacity. Incremental stroke lengths may be obtained.



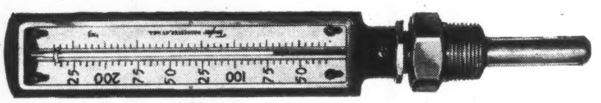
Hydraulic Remote Control

HAVING a total weight exclusive of tubing of only 3.7 pounds, a hydraulic remote control system has been announced by Sperry Products Inc., 1505 Willow Ave., Hoboken, N. J. Retaining all the advantages of the previous larger units in that only a single flexible copper tube is used, the system is completely enclosed. Installation is easily accomplished with three bolts for each of the two pieces that make up the unit. Made of bronze throughout, any motion of the transmitter arm will be duplicated by the receiver arm. Either arm will move through an arc of 60 degrees. The receiver arm may be drilled at any location to obtain the desired linear travel of the actuating rod. Recommended for position indicator controls and for such applications as throttle, mixture or governor adjustment, the system is dust and water-proof. Connecting tubing may be of 35-foot length. Greater length is feasible under ideal conditions of temperature or load.



Industrial Thermometer

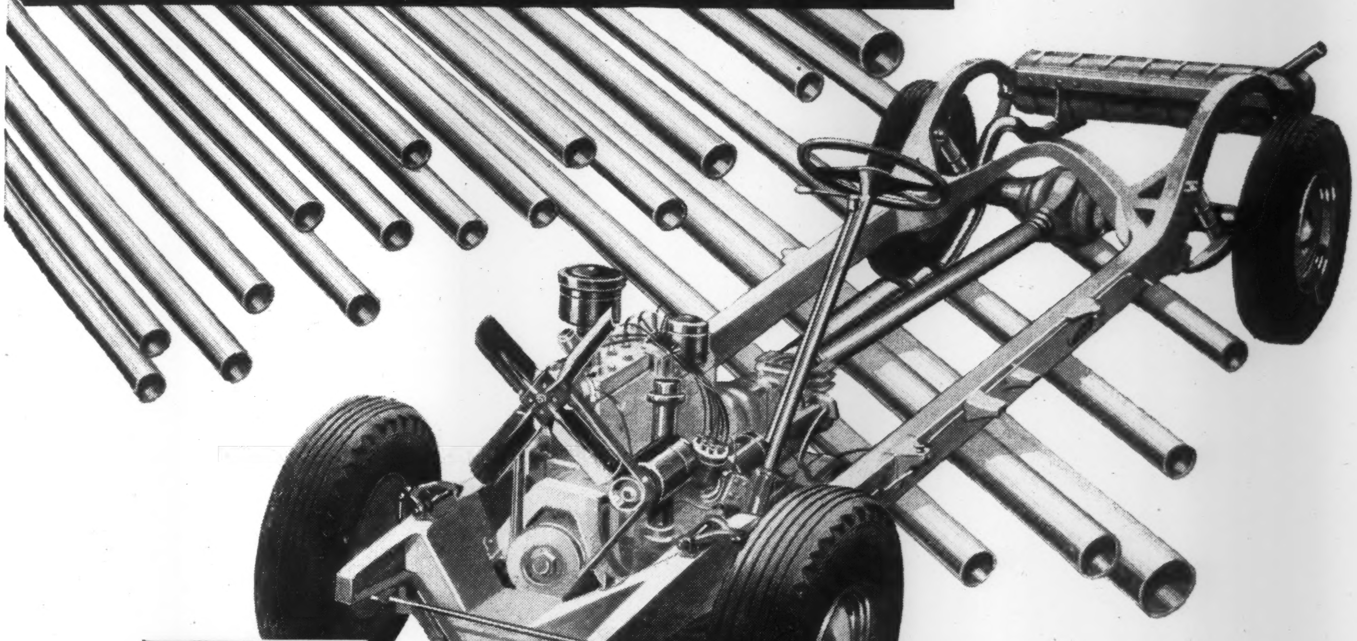
VERSATILE 5-inch mercury-in-glass thermometers have been made available for industrial use by Taylor Instrument Companies, Rochester 1, N. Y. Designed for use where space is limited and instruments are subjected to unusual vibration, the thermometer can be used for applications such as diesel engines, air compressors, lubrication oil lines on motors, generators, reduction gears, and small diameter brine, water and steam lines. Easily read tubing is encased in a compact, one-piece case, with a variety of stem forms: Straight, 90-degree angle, 90-degree right side angle, 90-degree left side angle, or 45-degree reclined (135-degree oblique) angle. Ranges available are: Minus 40 to + 110 F., 30 to 180 F. 30 to 240 F, and 200 to 500 F.



Centralized Lubrication

PERMITTING the lubrication of over a hundred bearings from a single pump circuit, the "M" system of centralized lubrication has been announced by the Trabon Corp., 1814 E. 40th St., Cleveland. Operated either manually or automatically, the lubricant, either oil or grease, flows only in

FOR STRONG, LIGHTWEIGHT PARTS IN AUTOMOBILES AND MACHINERY USE J&L ELECTRICWELD TUBING



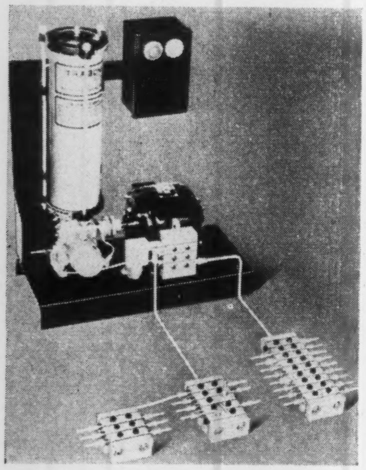
J&L STEEL

Weight reduction without sacrificing strength is possible in your products through use of J&L Electricweld Tubing. Many design engineers have discovered the economy of specifying J&L tubing for supporting members as well as for use in machine parts and formed sections. They also specify tubing for parts under dynamic loading for it will carry more load than any other section of the same weight. Write today for further informatior

JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PENNSYLVANIA

one direction from the pump to the master feeder. The master feeder measures and distributes the lubricant to subsidiary feeders located at convenient points on the machine. Each of the subsidiary feeders, in turn measures and distributes its flow of lubricant to the two to twenty bearings it serves. Action of the system is positive, each measuring piston operating progressively, forcing a measured charge of grease into its bearing line before the main flow of grease can proceed to the next measuring piston. A variety of feeder section sizes are available.

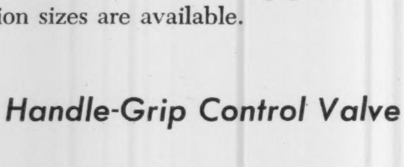


Handle-Grip Control Valve

MANUFACTURED BY Pneumatics Inc., Plymouth, Ind., a new handle-grip control valve is the newest addition to the company's ½-inch, 4-way control series. It contains a detention spring, permitting control to be fixed in either position. This compact valve is slightly over 5 inches high, 4 inches deep, and 3 inches wide. While available with a handle-grip control, this speedy valve is also furnished with the same body but with cam, palm, foot, pilot or solenoid controls.

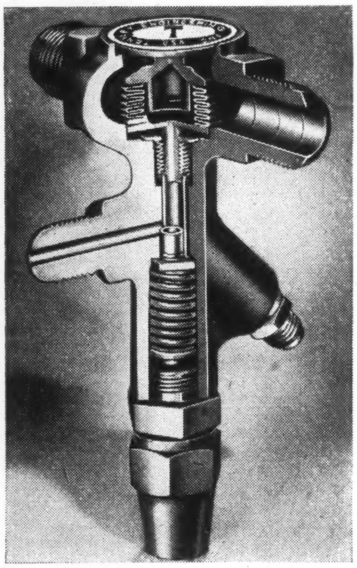
Speed-Control Valves

CONTROL of cylinder piston speed in both directions is accomplished by the two-direction speed-control valve manufactured by the Hanna Engineering Works, 1765 Elyston Ave., Chicago 22. Adjustable, the valve provides for control of inflow and outflow of air or oil independently to and from one side of a piston. One valve is used for oil operation, two for air. A spring-loaded, throttling type of valve, flow adjustment is possible by means of a screw adjusted stop located in surfaces of the valve at right angles to fluid flow. Ratings are 250 psi for air, 1000 psi for hydraulic systems. Pipe sizes available are ¼ to one inch in 5 steps.



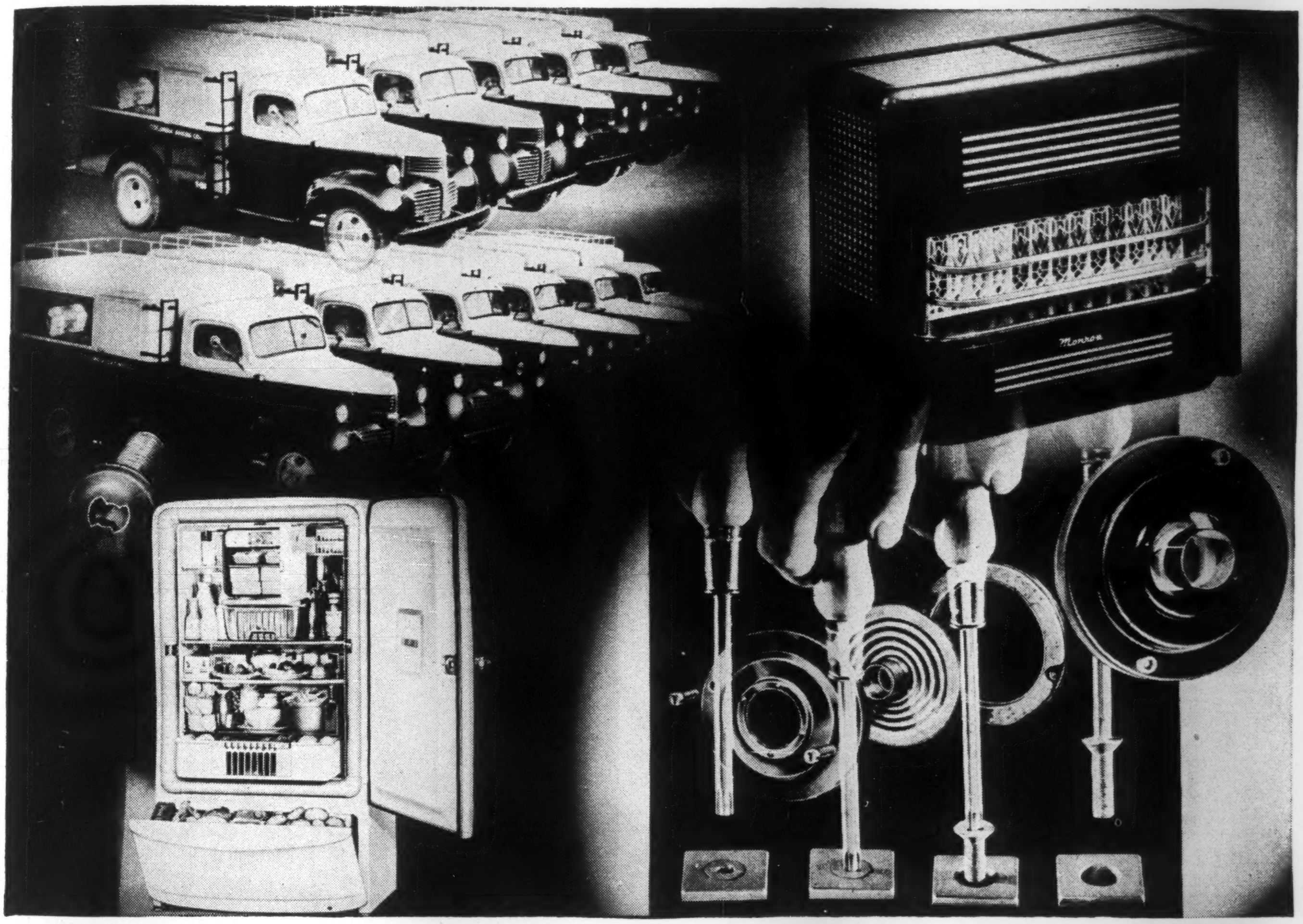
Thermostatic Expansion Valves

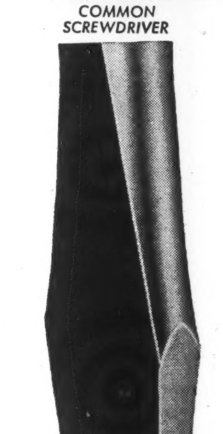
THERMOSTATIC expansion valves for regulating refrigerant flow into an evaporator have been designed by Tenney Engineering Inc., 26 Avenue B, Newark 5, N. J., for standard commercial use such as air conditioning, refrigerator boxes, etc. Valves are not affected by box temperature, entering warm air, or warm suction lines, and respond instantly to changes in suction vapor conditions. Close superheat control (such as 5 F superheat with plus or minus ½ F) can be maintained. Need for special charging and cross-charging to assure operation in a specific temperature range are eliminated. The valves are particularly adaptable to modern evaporators with forced air, small tubes, short passes and distributor header combinations, as well as for small evaporators or modern close-coupled coil and machine combinations.

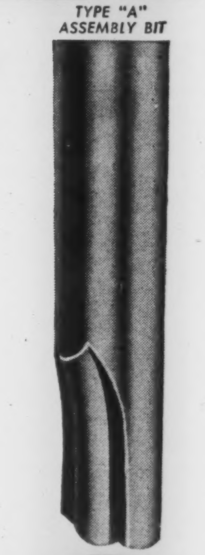


Flexible Bearings

MOUNTED IN RUBBER, Torflex bearings manufactured by the Harris Products Co., 634 Huron Road, Cleveland, are said to reduce the problems of noise, vibration, impact, and lubrication. The units consist of a tube of rubber stretched longitudinally between two concentric metal sleeves which prevent the rubber from returning to its original state. Pressure exerted by the rubber on the sleeves insures a high capacity mechanical bond between the metal and the rubber. The bearings will transmit torque, compensate for parallel and angular misalignment, and require no lubrication. They come in a wide range of sizes, are simple and easy to incorporate into designs and to install.







Clutch Heads Will Cut *Your* Assembly Costs Too

Just four typical examples telling what you may expect to gain by adopting the most modern screw on the market today... in lowered costs, safer, faster, smoother operation; plus simplified service.

The Lindsay Corporation, CHICAGO—"SIX instead of FIVE truck body assemblies"... this increase in production as compared with the use of other recessed head screws.

Dearborn-Monroe Company, CHICAGO—"Change-over to CLUTCH HEADS (*from other recessed head screws*) cut our Gas Heater assembly costs 22%... Your Type "A" Bit outlasts other bits 5 to 1... easy field service with flat blade is an important advantage."

Norge Refrigerators report—"Stepped-up production and elimination of cabinet damage"... tribute to CLUTCH HEAD'S protection against driver slippage.

Quam-Nichols Company, CHICAGO—("Adjust-a-Cone" Speaker Units)—"A 20% increase in assembly operation... damage from driver slippage went to zero... service problems simplified by screwdriver operation."

These and other exclusive CLUTCH HEAD *advantages are available for every type of assembly line production. Investigate and convince yourself. Send for package assortment of* CLUTCH HEAD *Screws, sample of Type "A" Bit, and illustrated Brochure.*

CLUTCH HEAD SCREWS

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

L. R. Burr

H. D. Kelsey

John Elmer Housley

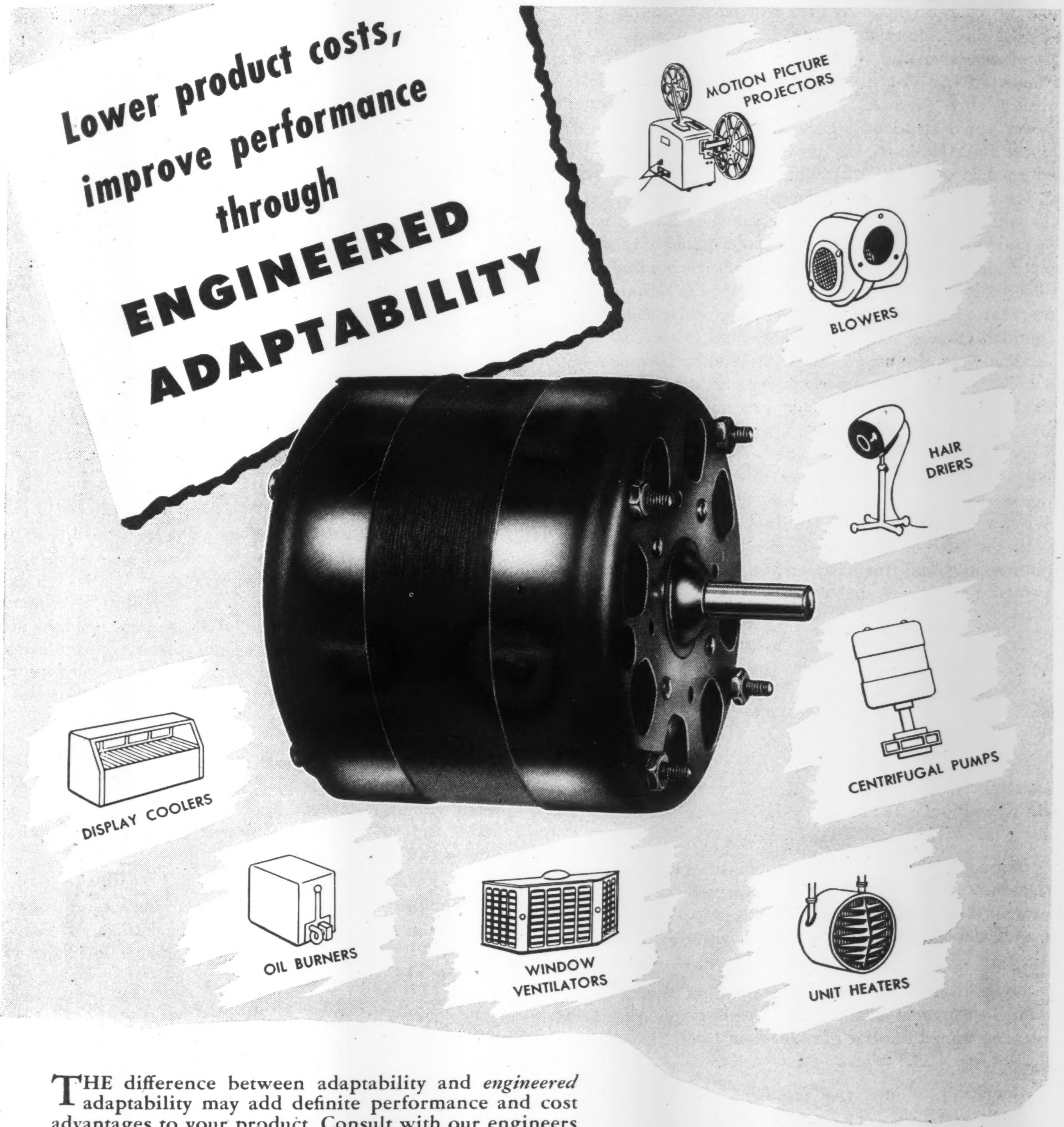
MEN... of... machines

L R. BURR, the new chief engineer of Kold-Hold Mfg. Co., has had considerable experience in the air-conditioning field which will assist him in capably fulfilling his duties. Following his graduation with liberal arts and mechanical engineering degrees, Mr. Kelsey became associated with General Refrigeration Corp. as development engineer. He remained here for three years, on development and test work in connection with air conditioners used by airlines for cooling and heating planes at airports, as well as on design of self-contained air-conditioning equipment and reciprocating compressors. He then joined The Trane Co., where as design and development engineer, he was responsible for bus, truck and railroad car air-conditioning unit design. His work also included the design and development of the company's Turbovacuum compressor, brazed aluminum plates for intercoolers on fighter planes and bombers, and welded stainless steel plates for exhaust gas heat exchangers for aircraft de-icing. Mr. Burr had been connected with The Trane Co. for five years before becoming chief engineer with Kold-Hold Mfg. Co.

H D. KELSEY has recently been appointed managing engineer of General Electric company's new aircraft gas turbine division, which was organized from various departments engaged in developing and manufacturing gas turbines. Mr. Kelsey joined General Electric in 1920 after graduating from Syracuse university. Following completion of the company's test course, he was assigned to the turbine engineering department as engineer. In 1929 he was named engineer in charge of the centrifugal compressor department. Later he served in a similar capacity in the mechanical drive section of the turbine department, and then was placed in charge of the air conditioning department. For three years he headed the engineering sections of both the air-conditioning and commercial refrigeration departments, and in 1942 was commissioned to establish a new turbosupercharger factory for the company at Fort Wayne, Ind. Two years later in the capacity of works manager he supervised all engineering work at the plant. Now as managing engineer of the aircraft gas turbine division, Mr. Kelsey is in charge of both engineering and manufacturing of turbines.

JOHN ELMER HOUSLEY, district power manager, Aluminum Co. of America, has been nominated for the presidency of the American Institute of Electrical Engineers. Since graduating from the University of Tennessee with a bach-

Lower product costs, improve performance through ENGINEERED ADAPTABILITY


MOTION PICTURE PROJECTORS
BLOWERS
HAIR DRIERS
CENTRIFUGAL PUMPS
DISPLAY COOLERS
OIL BURNERS
WINDOW VENTILATORS
UNIT HEATERS

THE difference between adaptability and *engineered* adaptability may add definite performance and cost advantages to your product. Consult with our engineers NOW if you plan to use production-run quantities of small motors (1/500 to 1/15 H.P.) in your 1947 products. A skilled experienced factory engineer will meet with your engineers in your own plant. He'll make a thorough study of your product design and probably recommend a motor of standard type. *But,* he'll *engineer* the physical and electrical characteristics of the motor—varying them to adapt the motor *specifically* to your product. Write today, describing briefly your product design plans.

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FASCO

Pilot
Fractional H.P.
MOTORS & BLOWERS

elor of science degree in electrical engineering in 1915, he has been associated with various branches of the Aluminum company. From 1915 to 1922 he was with the Aluminum Ore Co., a subsidiary, as apprentice engineer, electric shop foreman, electrical engineer and assistant power superintendent. During 1922 he was sales engineer for Aluminum Co. of America, and in 1923 was engaged in extensive rehabilitation of 1200 electric motor drives and the industrial electric distribution system of the Aluminum Ore Co. In this connection he was instrumental in persuading the electric motor industry to change motor designs to withstand severe duty in chemical fields. Silver soldering of squirrel-cage rotors was introduced in the field on a large scale, and other improvements in distribution circuits, substation design, and motor control insulation were also made. Early in 1924 he became assistant district electrical superintendent and in 1927 district electrical superintendent, later superintendent of power and district power manager, the position he now holds. Since 1927 Mr. Housley has been in charge of engineering, operation and maintenance of the hydro-electric power system of the company in Eastern Tennessee and Western North Carolina. He has been actively identified with the problems of interconnected power system operation, load and frequency control, telemetering and carrier current relaying and communications.

◆

Thomas J. Herrick is now assistant professor of aeronautical engineering, Purdue University. Formerly he had been research engineer of McDonnell Aircraft Corp., St. Louis.

◆

Paul R. Guerin has been promoted to the position of assistant chief engineer by Towmotor Corp., Cleveland. He was production engineer with the same firm.

◆

Bernard J. Wolfe has severed his connection with Bausch & Lomb Optical Co. to become a designing engineer in the experimental engineering department of Eastman Kodak Co., Camera Works, Rochester, N. Y.

◆

James R. Downing, who since 1942 has been connected with development work on the atomic bomb project, has been appointed director of research at Cook Electric Co.

◆

Gordon J. Berry, vice president of Electric Products Co., Cleveland, has been elected president of the Electric Industrial Truck Association for 1946.

◆

E. D. Flintermann, the new president of the Steel Founders' Society of America, is connected with Michigan Steel Casting Co., Detroit.

◆

Col. C. E. Davies, secretary of the American Society of Mechanical Engineers, was recently awarded the medal of the Legion of Merit for meritorious service while on active duty with the Ordnance Department of the United States Army. Colonel Davies has resumed his duties with the Society, after an extended leave of absence.

◆

Seth H. Stoner has recently been appointed assistant chief engineer in charge of automotive and tractor applications, New Departure Division of General Motors Corp. Since June, 1941 Mr. Stoner has been on leave with the Navy and previous to that time was executive engineer for the New Departure Division.

◆

George D. Clark, formerly chief motor engineer, Packard Electric Division of General Motors, has been named chief engineer of Russell Electric Co.

◆

James C. Barnaby has been transferred to the general engineering staff of Worthington Pump & Machinery Corp., Harrison Works, as assistant director of research and development. He had been consulting engineer.

◆

George H. Kublin has returned to the General Motors Overseas Operations as chief engineer of product engineering, Detroit office. Formerly assistant chief engineer of the overseas organization, Mr. Kublin served as chief engineer for the Fisher Tank Division of General Motors during the war.

◆

William E. Bradley has succeeded **David B. Smith** as director of research of the Philco Corp. Mr. Smith was recently named vice president in charge of engineering. Mr. Bradley joined the Philco organization in 1936 upon graduation and five years ago, after serving in various capacities, he was placed in charge of the advanced research section. Early in 1945 he became assistant director of that division, the position he held prior to his present appointment.

◆

Paul F. G. Holst, who during the past 16 years played an active part in the development of audio, radio and television circuits, has been appointed engineer in charge of audio and television development in the manufacturing division of The Crosley Corp.

◆

William D. Hazlett recently became executive engineer of the Aeroproducts Division, General Motors Corp., Dayton, O. He had been section engineer in charge of drafting and standards.

◆

Henry H. Hering Jr., formerly model test engineer, Dodge Division of the Chrysler Corp., Chicago, has joined General Electric X-Ray Corp., Chicago, as mechanical design engineer.

◆

Robert V. Kerley, formerly a mechanical engineer with the USAAF, Air Technical Service Command, Wright Field, has been made director of aeronautical research of the Ethyl Corp., Research Laboratories, Detroit.

◆

Arthur W. Harris is the new engine production engineer, Chevrolet Motor division, Detroit. Previously he had been assistant chief engineer of the Chevrolet Aviation Engine division, Buffalo.

◆

Sidney Oldberg has joined Wilcox-Rich Division of Eaton Mfg. Co., Detroit as executive engineer. For ten years before this appointment, Mr. Oldberg was with Chrysler Corp. most recently in charge of aircraft engine research and development.

How to Reduce 25 Lubrication Points to ONE!

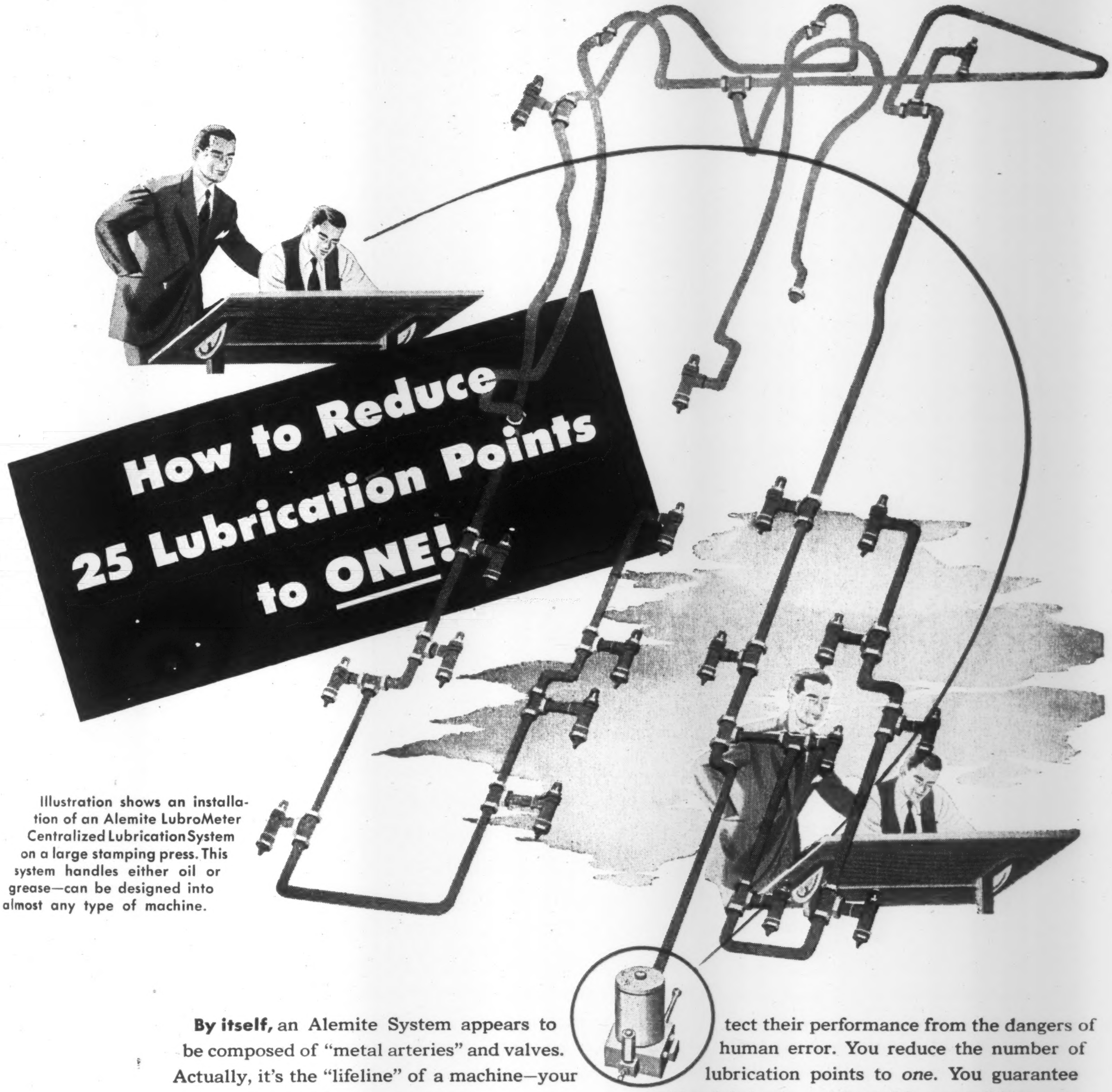

Illustration shows an installation of an Alemite LubroMeter Centralized Lubrication System on a large stamping press. This system handles either oil or grease—can be designed into almost any type of machine.

By itself, an Alemite System appears to be composed of "metal arteries" and valves. Actually, it's the "lifeline" of a machine—your machine, if you like.

Maybe your machine has 10 lubrication points —or 25, or 40, or 60. With an Alemite System there'd only be *one*—the one central point where the lubricant is introduced. From that *one* point *all* points would be positively lubricated with a metered quantity of grease or oil without *stopping the machine!*

Results like these are common. A huge rotogravure printing press had to be shut down 2 hours daily for lubrication, and bearing failures were frequent. Today, with an Alemite Centralized System, the press is positively lubricated, while in operation, in only 10 minutes per 24 hour day. *And*—bearing failures have ceased to exist.

When you design Alemite Centralized Lubrication into your machines you automatically protect their performance from the dangers of human error. You reduce the number of lubrication points to *one*. You guarantee positive lubrication without "time-outs." You cut repairs to a minimum . . . lengthen machine life . . . increase satisfaction. It all adds up to More Productive Time per machine. And that enhances your reputation as a designer.

* * *

Without obligation, have an Alemite Lubrication Specialist demonstrate one or all 4 new Centralized Systems right at your desk with transparent working models. Also, ask for any technical help you want regarding Alemite Systems. Drop a note on your letterhead to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

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Another Product of
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Only Alemite Combines ALL 3 in Lubrication

1. EQUIPMENT 2. PROCEDURES 3. LUBRICANTS

Design Abstracts

Kettering on Standards

SOMETIMES when you make a standard, you set up things that rule other things out. We have gone through one in connection with our diesel locomotives that has been very interesting.

We started out with certain types of pistons and rings. Everybody seemed to be perfectly happy to change them at sixty or seventy thousand miles. But I was very unhappy that they were happy about it, so I said, "Well, why do you think that is good?"

"Well," they said, "Sixty-five or seventy thousand miles is quite a distance, and that is about all you can expect a piston to do."

I said, "Why?"

"Well," they said, "Just figure how far that piston ring has to travel up and down."

I think it worked out that it travels almost as far as the train does. But I said, "What has that got to do with it?"

They said, "Well, if you were dragged sixty thousand miles over a surface, maybe you would want to be changed, too."

So we tried to find out some things and we discovered a mystery. Where the top piston stopped it always eroded, and the cylinders would wear a little groove where the top ring stopped. I can show you a stack of papers four feet high on all the theories in the world as to why that is.

We said, "Well, let's see if we can measure something in there."

So we finally did. We found out the reason it eroded was because it got too hot. No one had ever measured the temperature because, although it was a fairly simple thing, it was rather difficult to do. So after we discovered heat was the cause of the erosion, we developed an entirely new type of piston and a new type of ring. We have records of the new type rings that have run 750,000 miles. All of our pistons are discarded at one-and-a-half million miles. I don't know why we do that. They seem to be as good as when they started out, but that distance is beyond the limits the experts think they should go.

Now here is an amazing thing. We took one of the old type pistons that had run seventy-five thousand miles before it had to be changed, and sawed it in two, including the rings. We also sawed a new type piston that will run a million-and-a-half miles. To make things interesting we set a card in front of each piston. The card in front of the old type piston read, "This aluminum piston cost $100." The card in front of the new type steel piston read, "$50."

Everyone that looked at the display said, "Well, that $50 piston is no good. I wouldn't use that piston if you gave it to me." Then we showed them, on the back of the card, the performance record of each.

The fellows said, "You can't tell me that, because I can tell by looking at them that this $50 one is no good."

I said, "How do you know by looking at a piston whether it is any good or not? Were you ever a piston in a diesel engine?"

It was not a question at all of whether the piston was aluminum or steel or this or that. It was a question of taking a set of standards that had been developed in one case and trying to apply them in another. So in spite of standards, you have to be questioning all the time.—*From an address by C. F. Kettering, delivered at a recent ASTM Detroit district meeting.*

Propellers versus Jet Propulsion

DESPITE the spectacular development and achievements of turbo-jet power plants in aircraft in recent months, the conventional propeller, particularly when combined with the gas turbine engine, is still superior and likely to remain so for some time to come, according to a study of the relative performance of the two types of plants for equal fuel consumption at maximum rating.

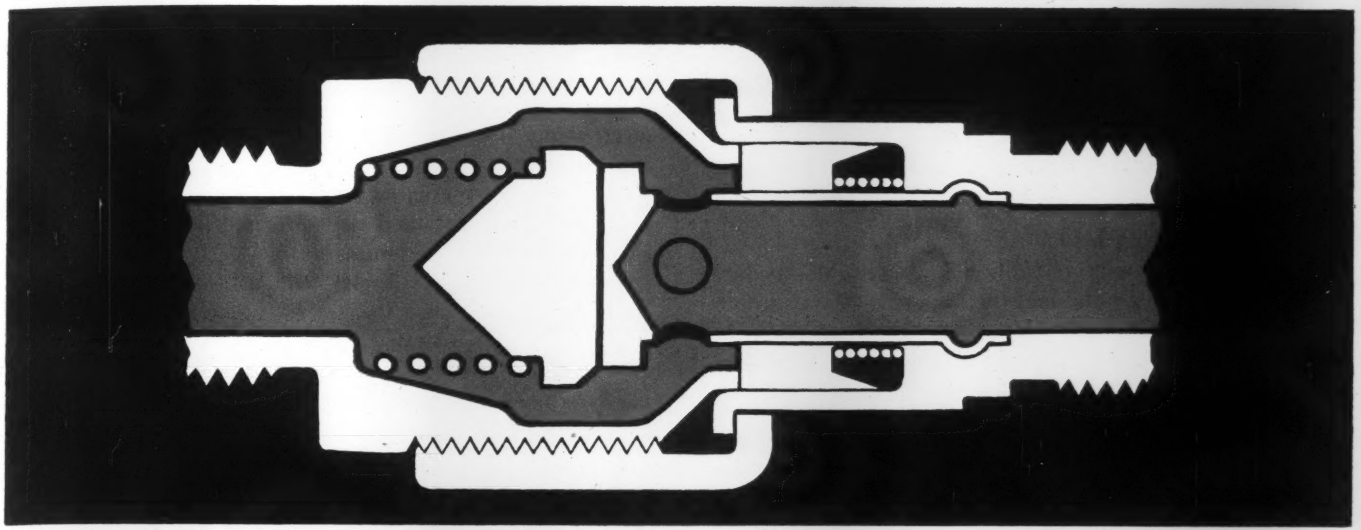
It might be argued that probable future improvements in fuel economy of turbojets will make the present comparison invalid. This is not believed a proper objection, however, inasmuch as any improvement in efficiency due to internal improvements in the turbojet will be equally applicable to the propeller turbine.

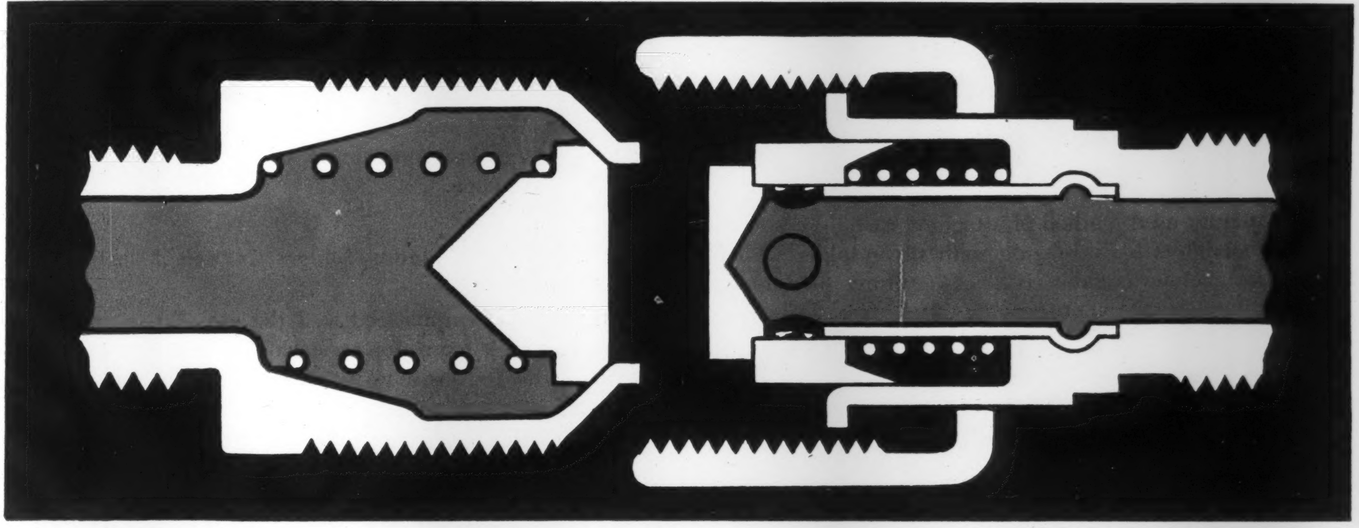
Many predictions were made several years ago both within and without the propeller industry that catastrophic reductions in propeller efficiency would be experienced at speeds of about 400 mph. At a later date when fighter planes had exceeded 400 mph the critical speed was revised to 500 mph.

To date airplanes with propellers have attained level flight speeds of slightly over 500 mph. No major reductions in propeller efficiency have been experienced. With present-day knowledge of propeller design there is every reason to believe that quite reasonable efficiency can be expected at speeds in the 600-700 mph range.

It has been found, however, that blade sections should be thin and that well-faired shanks are necessary, but otherwise no particular design changes have been made. It may be concluded from this experience that propellers for gas turbines on airplanes in the 400-500 mph category need not be appreciably different from those for piston engines, so far as aerodynamics are concerned.

For higher speeds, of the order of 600 mph, the problem changes appreciably, and new propeller development will be necessary along the lines of thinner blade sections, the application of the sweepback principle to propeller blades, and perhaps the use of a propeller type having eight or ten blades of relatively small diameter in which only the working part of the blades is exposed.—*From a paper by George W. Brady, chief engineer, Curtiss-Wright Corp., Propeller Div., presented at the recent national aircraft propulsion meeting of the Institute of Aeronautical Sciences.*





AEROQUIP Self-Sealing Couplings *

eliminates the use of 2 Valves

With the Aeroquip Self-Sealing Couplings, fluid carrying lines may be instantly disconnected without leakage. Upon separation each half of the coupling automatically closes the line or tube to which it is connected and positively retains the fluid carried therein. No leakage or loss of fluid occurs and no air is allowed to enter the system on reconnection. Suitable for hydraulic and refrigeration systems as well as any lines carrying oils, fuels, coolants, or refrigeration gases.

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Detachable Hose Fittings
Hydrofuse
X-Washers
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Cemented Carbides

(Concluded from Page 138)

the shoulder by means of a set screw or threaded ring is satisfactory and often can be substituted for the conventional arrangement of set screw at right angles to the surface of the part being held.

Threading of carbide generally is not advisable because the thread must be ground and the high cost of such grinding is seldom warranted. However, blind or through-threaded holes in carbide parts can be made either by brazing threaded inserts into holes in the carbide or filling the holes with brazed-in material which then can be drilled and tapped. Of course the holes in the carbide must be drilled before sintering.

Using shrink or press fits of steel around carbide frequently is practical provided operating temperatures are not high. Where a wide range of service temperatures is involved the steel will, of course, expand more rapidly than the carbide and thus loosen the shrink or press-fit grip. Shrink or press fits of carbide over steel should not be attempted unless the section of carbide is very heavy because, while carbide is stronger than steel in compression, it is weaker than steel in tension.

A frequently used method of attaching carbide inserts to machine members is by brazing with silver solder. Generally the machine member is steel requiring medium or high hardness in conjunction with the wear resistance of the carbide on critical surfaces. Customary brazing procedure after heat treatment of the steel draws the steel hardness back considerably and one of several methods may be used to avoid this difficulty. The steel part may be made of a high-speed type of steel which hardens at high temperatures, permitting brazing at a temperature of about 1200 F without materially reducing the hardness of the steel part. Another common method involves using an air-hardening tool steel, heating the entire steel part during brazing with a medium or high-temperature braze, air cooling the entire unit to room temperature, and then drawing the assembly to develop the desired hardness in the steel.

In selecting proportions of steel and carbide for brazed assemblies, a sufficient amount of steel should be used adjacent to the braze. A thin strip of steel brazed to a piece of carbide frequently will cause checks or cracks during cooling, finishing, or use, due to thermal strains set up in the union. When long thin sections of carbide are to be attached to a steel backing, mechanical mounting is preferred because high thermal strains are set up when brazed construction is used. If mechanical mounting is impractical, a series of smaller pieces of carbide brazed end to end will result in a minimum of thermal strain.

For carbide bushings a wall thickness approximately 15 to 25 per cent of the diameter usually is most economical. A thin-wall bushing generally is more expensive, as the difficulty of manufacturing a thin-wall bushing increases cost more rapidly than savings of material decrease it.

On thin strips of flat carbide, a proportion of length to thickness between 8 to 1 and 12 to 1 usually is most economical. Beyond that, the cost of manufacture increases faster than savings of material decrease the price.

Other methods of attaching carbide units to various materials are: Cementing with plastic resin, soft soldering, and bonding with a low-melting-point alloy. Plastic resin (such as Vinyl resin) is used where a material cannot be brazed or is impractical to braze. Applied all around the insert under heat and pressure, such resin-cemented bonds have strength on the order of an average solder. Of course the bond created by soft solder is not equal in strength to that obtained with brazing, nor will it tolerate as high an operating temperature.

Bonding of carbide inserts with a low-melting-point alloy is used where even the low heat required for soft soldering cannot be tolerated. Since the bond is purely mechanical, the parts involved should have knurled or roughened surfaces to provide adequate anchorage and a space of at least 1/32-inch provided between them for pouring in the alloy, which melts at about 250 F.

CORROSION RESISTANCE: The corrosion resistance of cemented carbide compositions is largely determined by the binder or matrix metal since the carbides themselves are extremely inert. Cobalt, a commonly used binder, has good resistance, comparable under some conditions to the best corrosion-resistant materials such as nickel, Monel and the Hastelloys at room temperature. For use in solutions at higher temperatures, low cobalt-content grades have the best resistance.

Corrosion Resistance Seldom Controls Selection

It should be emphasized that the use of cemented carbides for their corrosion resistance alone is practically never justified since more workable, more resistant, lower cost materials are available. However, when wear also is involved the combined corrosion and wear resistance of carbides often will give better service. The following summary is based on laboratory tests* and thus is only indicative of the results that may be expected in service.

ACIDS: All grades are satisfactory in hydrochloric acid up to about 20 per cent acid by weight (50 per cent of concentrated acid). In nitric acid, the resistance is more erratic, the lower cobalt steel-cutting grades being best. Some grades appear satisfactory up to full strength at room temperature while other grades are unsatisfactory even at 50 per cent strength. In sulphuric acid, results thus far indicate carbides are satisfactory only in low strengths (up to 15 per cent at room temperature).

ALKALIES: Carbides are highly resistant to all strengths of sodium hydroxide solutions at all temperatures up to boiling. It has not been determined whether there is any caustic embrittlement over long periods of time, but this appears unlikely. Resistance to ammonia and other alkalies should be excellent.

SALT AND MISCELLANEOUS CORROSIVES: Salt solutions up to 25 per cent by weight have only a slight effect and results should not be any different at higher concentrations. Some laboratory tests in various solutions encountered in rayon spinning processes have shown virtually no attack takes place except in the complex sulphuric-acid-base spinning baths where attack is moderate and service life should be satisfactory.

Collaboration of the following organizations in the preparation of this article is acknowledged with appreciation:
Carboloy Company, Inc. (*Figs.* 1, 2, 3, and 7) ...... Detroit
Kennametal, Inc. (*Figs.* 4, 5, 6, 8,) .... Latrobe, Pa.

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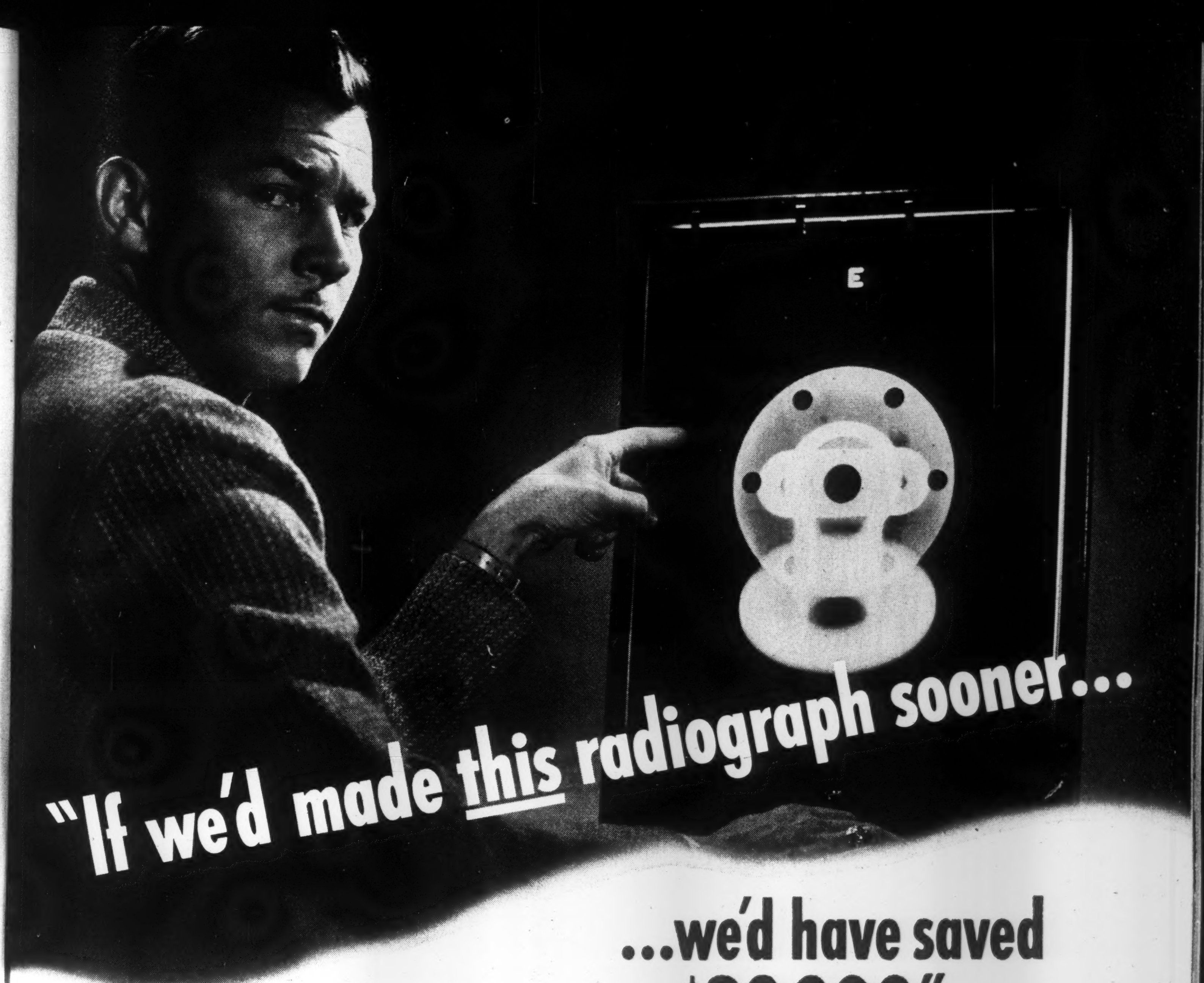
* Tests conducted at laboratories of Kennametal Inc.


"If we'd made this radiograph sooner...
...we'd have saved $30,000"
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A manufacturer of valves, whose foundry technic seemed "okay," decided pilot radiographs weren't needed. But when machining was done and the valves assembled, enough were found defective to cause a loss of many thousands of dollars.

Radiography, put to work at the right time and place, pays off. Men who have figured *cost-wise* what it can do for them . . . to improve design, speed production, and lower costs . . . make full use of x-ray.

Radiographs show your engineers where to correct faulty design . . . how to reduce weight safely . . . how to cut costs at many stages of manufacture . . . how to build in extra dependability.

Order-jammed foundries get a welcome production spurt when radiography shows how to get into sound casting production fast. High-value, high-volume machine shops operate at rock-bottom cost when radiographic inspection keeps internally unsound castings off the production line.

Welding gains acceptance . . . new markets . . . higher volume . . . because radiographs prove weldments sound.

And these are only a few high spots in radiography's list of industrial functions. You can find more—if you look for them—right in your own plant. Why not get together with your radiographer or the local x-ray dealer and see if you are missing any chances to make radiography pay extra dividends? Or write to

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You can profit by our EXPERIENCE in building

2½ MILLION BRIGGS & STRATTON ENGINES

ONLY with Briggs & Stratton 4-cycle engines can you profit by the skill and experience of an organization which has built 2½ Million air-cooled engines. *Their record of dependable performance* through the years is your assurance that "it's powered right when it's powered by Briggs & Stratton"—a factor of utmost importance to every user, dealer and manufacturer of gasoline powered appliances, farm machinery or industrial equipment.

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Milwaukee 1, Wisconsin

4 CYCLE
BRIGGS & STRATTON
GASOLINE ENGINES

Air-Cooled Power

Front and Rear Engine Autos

(Concluded from Page 148)

the Kaiser car, the taking out of six bolts, after disconnecting the controls, permits removal of the power plant—a splendid feature for periodic servicing. However, if it is desired to remove only a small part or subassembly, there would not be accessibility as we know it today. (Heaven forbid! It is bad enough as it is, and it could be bettered if vehicles were designed for easy servicing and not solely for quick assembly on the conveyor lines.) Such removal probably would require the dismantling of an adjacent part or parts. In many cases it would be easier to take out the whole unit and do the work on a stand but there would still be the problem of getting at the part if it is not on the outside or at an end.

Conclusion: It will be seen that there are many advantages and disadvantages with each of the systems discussed and, as in all engineering problems, the final design is a compromise. All the ideals of each type cannot be combined into one system.

To complete the picture on power plants and drives, it may be pointed out that the White and ACF busses have horizontal engines located below the vehicle floor. This presents an ideal weight distribution setup. In this large type of vehicle, the floor can be kept fairly close to the ground and yet provide ample clearance space beneath for the power plant. Naturally accessibility is forfeited, calling for maintenance from the garage pits.

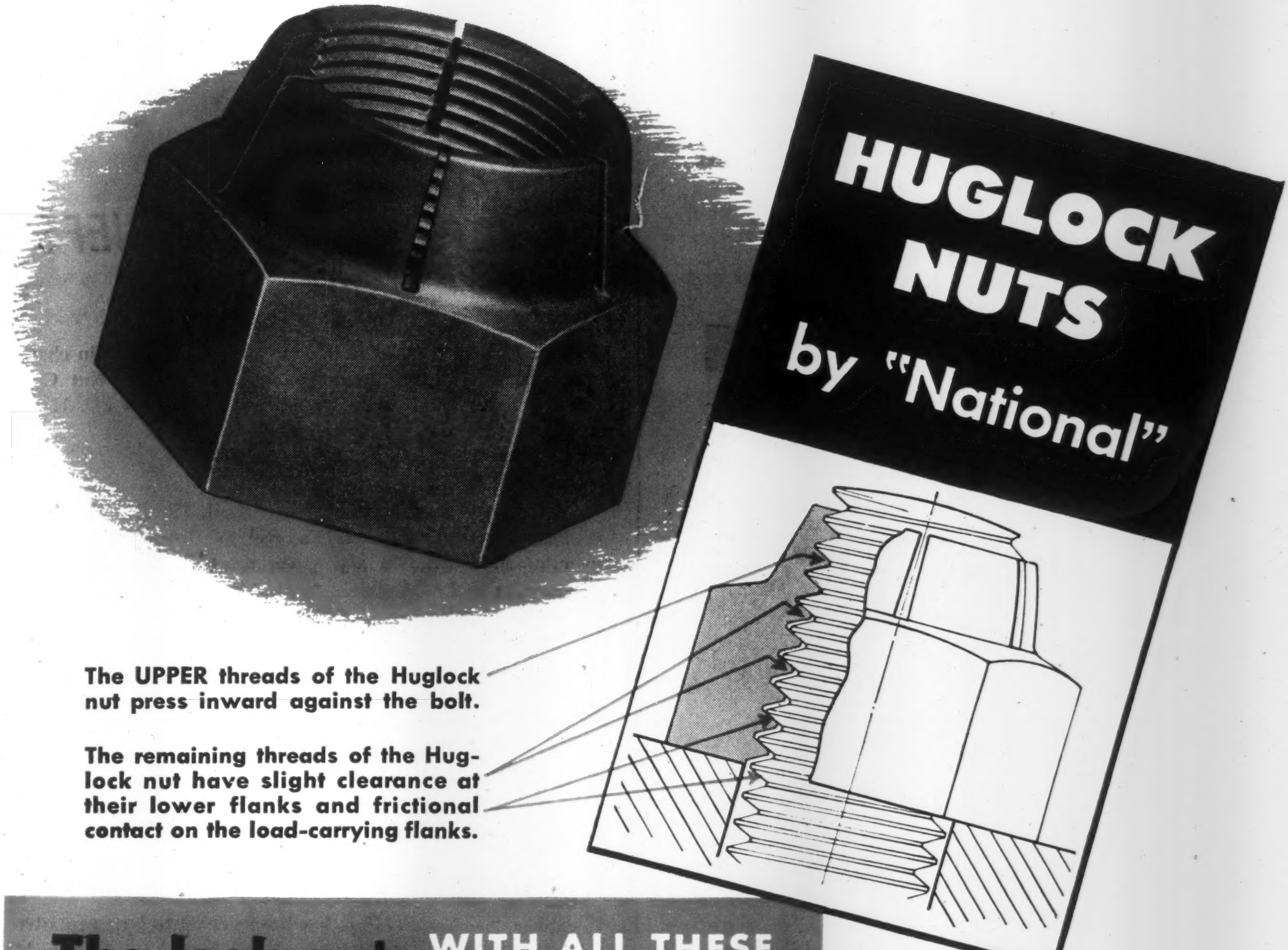
Flying Wing Is Latest Radical of the Skies

SCHEDULED to be flight-tested this fall at Muroc Army Air Field, California, the newly developed Flying Wing—or XB-35—has no conventional fuselage, its entire operating mechanism being housed within the wing.

This radical plane is, as the name implies, just a wing. To insure minimum drag, construction follows the swept-back wing principle, general contour of the wing being a wide "V". Overall length of the plane is slightly over 48 feet. Its width, from tip to tip, is approximately 172 feet, or about 31 feet greater than the span of the B-29. Thickest part of the wing, forward of center, is almost 7 feet and it is at this location that quarters for the crew of 15 are installed. Position of the pilot is in the center near the leading edge at the top of the wing, and the pilot is protected by a protruding blister similar to the types used in present-day fighter planes.

Power for the Flying Wing is provided by four Pratt & Whitney, X-Wasp R-4360 pusher-type engines, with contra-rotating propellers having reversible pitch for braking. The tricycle landing gear is retractable and has a dual main wheel and single nose wheel. Cabin pressurization is employed and both the oil and fuel lines are self-sealing.

One of the plane's many revolutionary features is the alternating current electric system, main advantages of which are improved compactness and maximum lightness in weight.

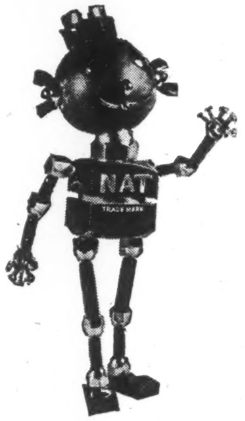


HUGLOCK NUTS by "National"

The UPPER threads of the Huglock nut press inward against the bolt.

The remaining threads of the Huglock nut have slight clearance at their lower flanks and frictional contact on the load-carrying flanks.

The lock nut WITH ALL THESE ADVANTAGES . . .



1 *One-piece construction*—reduces assembly time and avoids possible failure to install supplemental locking medium.

2 *Self-locking action*—the Huglock Nut grips the bolt threads and maintains locking effect, whether seated or not, on the load-carrying flanks of the threads.

3 *Axial thread play eliminated*—Huglock Nuts are vibration-proof and shock-proof.

4 *Thread load is distributed* over all the threads of the nut, instead of being concentrated on the few bottom threads.

5 *Maximum thread shear strength*—full strength of the most modern 180,000 lb. tensile alloy bolt will not strip threads in nut. (Compare this with other designs.)

6 *All metal construction* permits applications involving high temperatures, oil or other types of moisture. Self-locking is maintained at high temperatures.

7 *Easily installed*—Huglocks start freely and may be tightened rapidly by standard hand or speed wrenches.

8 *Economical to use*—cotter pins, lock washers, key plates, jam nuts or other secondary locking devices are completely eliminated.

9 *Re-usable*—repeated removal and re-use on the same bolt or a similar bolt will not destroy the locking action.

10 *Controlled preset torque values* are built into Huglocks, insuring the vibration-proof results visioned in your engineering designs.

U. S. Patents 2290270, 2333290, 2337797. Other Patents Pending

National
HEADED AND THREADED
PRODUCTS

Other "National" Specialties Include the Following:

Phillips Recessed Screws
Laminar Flow Screws
Marsden Lock Nuts
Dynamic Lock Nuts
Drake Lock Nuts
Place Bolts
Lok-Thred Studs
Rosán Locking System
Scrivets
Hi-Shear Rivets
Sems
Clutch Head Screws

Send for samples and consult with us on *any* fastener question.

THE NATIONAL SCREW & MFG. CO., CLEVELAND 4, O.

BRONZE BEARINGS

TO YOUR SPECIFICATIONS

When you buy bronze bearings you know what you want. The problem is to get them made to *your specifications* . . . of uniform quality as to alloy and machining . . . and get the right quantity delivered at the right time. For a quarter of a century Shook has been doing just that for manufacturers in many fields . . . aviation . . . automotive . . . industrial . . . marine . . . mining. These producers have looked to Shook for bronze bearings year after year because they got what they wanted . . . when they wanted it. Shook's production "know how," its war-enlarged foundries, machine shops and laboratories, its staff of field engineers and Shook's desire to be of lasting service are available to you.

SEND SHOOK YOUR BLUEPRINTS

You can easily find out that Shook can be of real service to you. Just send your blueprints and specifications to Shook. In short order you will receive Shook's quotation together with proof of Shook's ability to produce bearings of uniform quality in alloy and machining. Then you will also learn how Shook's extra measure of service has helped others and how it can help you too.

SB-114

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SHOOK BRONZE

CORPORATION

GREENLAWN AND LAKE STREETS • LIMA, OHIO

BUSINESS AND SALES BRIEFS

ELECTION of Charles E. Heintz as vice president in charge of sales has been announced by Elastic Stop Nut Corp. of America, Union, N. J. For the past two years Mr. Heintz served as general sales manager.

◆

Following the resignation of Joe E. Earll, The B. F. Goodrich Co. has named Charles H. Kanavel as district manager of the automotive, aviation and government sales division, with headquarters in Los Angeles.

◆

David Crawford has been appointed Mid-West district sales manager for The American Welding & Mfg. Co. of Warren, O., and will cover southern Ohio, Kentucky, Indiana, and a portion of Illinois. Before joining American Welding, Mr. Crawford was employed in the Bar Sales Division of Republic Steel Corp.

◆

According to an announcement by Allen-Bradley Co. of Milwaukee, the Boston office has been moved to larger quarters at 55 Oliver St. M. H. Hallenbeck, supported by Charles M. McCoombe, will remain in charge as district manager.

◆

Formerly connected with the West Penn Power Co. of Pittsburgh, Ralph H. Lightner has been made general sales manager of Titan Metal Mfg. Co., with headquarters in Bellefonte, Pa. He succeeds J. B. Craig, vice president, who has been appointed controller.

◆

Under the supervision of Gerald L. Britton, a sales and service office has been opened at 16 State St., Rochester, N. Y., by the Brown Instrument Co. division of Minneapolis-Honeywell Regulator Co. Also announced by the division are the appointments of O. J. Richardson and George W. Brown as industrial managers of the Detroit and Cincinnati offices, respectively.

◆

Consolidation of the Logansport, Ind., plant with the main plant in Baltimore has been announced by Gerotor May Corp. This involves the transfer of the entire Air & Hydraulics Division to the Baltimore plant.

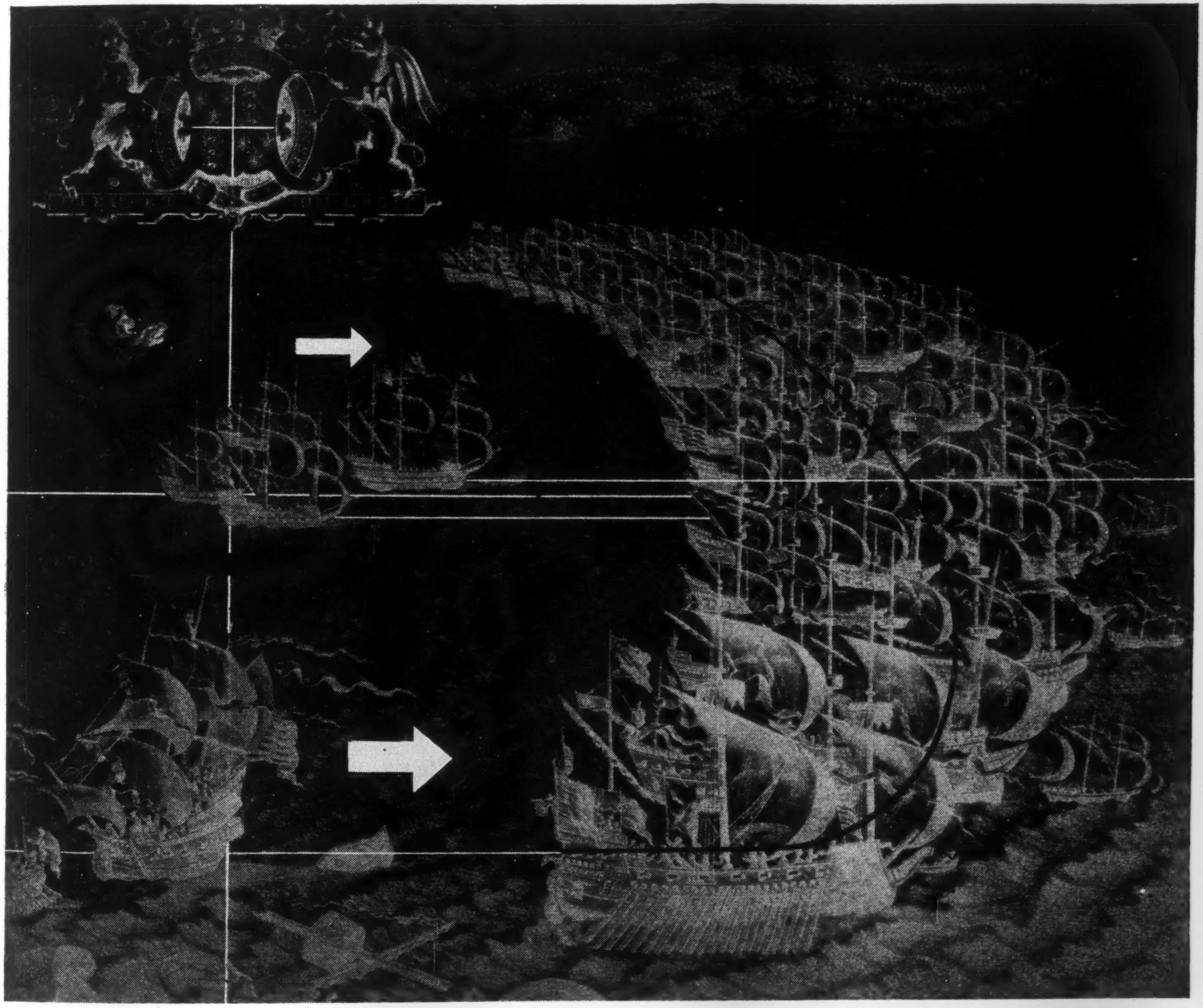
◆

Among recent appointments made by The Weatherhead Co., Cleveland, are the following: J. A. Strachan, sales manager of the Original Equipment division covering sales of all products to original equipment manufacturers in the refrigeration, air conditioning, railroad, marine, chemical, food and beverage processing, and dairy equipment fields; D. W. Holmes, sales manager of the Standard Parts division covering

A LITTLE DOES A LOT

The Spanish built their colossal Armada to smother England by sheer weight. But Drake, with a scratch fleet of nondescript ships, and some brains, reduced the Armada to an unpleasant memory.

Not only in warfare can a little do a lot, intelligently applied. Today a little molybdenum is doing a lot to improve the strength and toughness of good cast iron. The foundry man's market is broadened and many a user's production headaches relieved. Practical working data is available on request.



MOLYBDIC OXIDE—BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"
CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

Climax Molybdenum Company
500 Fifth Avenue • New York City

MOLY

Speed King of Assembling

Slotted or Recessed Heads

HOLTITE

"Thread Forming" SCREWS

If you are looking for speed-up methods in your assembling, eliminate time-consuming tapping operations by using HOLTITE "Thread-Forming" screws for metal-to-metal and plastic fastenings. Cutting their own threads in drilled, pierced or formed holes, these speed screws effect a stronger, tighter, vibration-resisting fastening as each thread stays tight in the perfect mating thread it has cut in the material.

Furnished in three types—Type "A", "Z" and "C." Send for information of specific uses and methods. Check your assemblies—you'll find these speed screws can be used to save time and strengthen many parts of your products.

Also furnished with Machine Screw threads

CONTINENTAL SCREW CO. New Bedford, Mass., U.S.A.

Specify HOLTITE

all products sold through distributor channels; and L. C. Doolittle, sales manager of the Aviation division covering sales of all products to original equipment manufacturers in the aviation, machine tool, metals processing, and machinery fields. All will make their headquarters in Cleveland.

◆

Following his recent release from the Army, Harold Henderson has returned to the Detroit sales office of The Formica Insulation Co. of Cincinnati. He will be associated with Frank Manley, who heads the Detroit office at 1342 Book Bldg.

◆

Kennametal Inc. of Latrobe, Pa., has established district offices at 538 North Erie St., Toledo, O., and in the American Bank Bldg., 600 Grant St., Pittsburgh. E. D. Porter, tool engineer, has been placed in charge of the former office. Fred J. Hennig Jr., assisted by P. R. Dinger, is manager of the latter office which will cover the western Pennsylvania and West Virginia territory as well as supervising field activities of offices in Erie, Rochester and Syracuse.

◆

Associated with The Bristol Co. since 1920, G. H. Gaites has been made district manager of the New York office located in the Fisk Bldg., 250 West 57th St. Prior to his appointment Mr. Gaites had been regional sales supervisor of the Cleveland and Pittsburgh territories.

◆

Recently discharged from the Army, Malcolm Wolcott has joined the staff of The Formica Insulation Co. of Cincinnati, manufacturer of laminated plastics. He will work in the Rochester, N. Y., territory under the direction of his father, E. M. Wolcott, who has been Rochester representative for the past thirty years. The Rochester office is located at 25 North St.

◆

Four new appointments have been announced by Ilg Electric Ventilating Co. Chicago. These are: F. H. Bigelow, manager of the office in Atlanta; H. H. Wilson, replacing Mr. Bigelow as manager of the Memphis, Tenn., office; E. Lloyd Widner, supervising the newly opened office in Knoxville, Tenn.; and Henry M. Lutes, manager of the office in Louisville.

◆

Previously chief engineer of the Thermex Division of the Girdler Corp., Paul D. Zottu has opened his own office and laboratory at 314 Washington St., Newton 58, Mass., where he will continue his consulting work in the field of industrial electronics.

◆

New Departure Division of General Motors Corp. has designated Jones Y. Pharr Jr., as southern representative for its ball bearings. With headquarters in Charlotte, S. C., Mr. Pharr will serve as sales engineer in North Carolina, South Carolina, Georgia and Alabama.

◆

To provide for expansion of company activities and the return of servicemen in the Specialties Sales Division of the industrial sales department, the National Oil Products Co. of Harrison, N. J., has appointed the following: Thomas J. Campbell, recently in the Armed Forces, to cover Long Island, Brooklyn and Queens, N. Y.; Daniel S. Rion, another ex-serviceman, to handle the South Carolina, Georgia and middle Tennessee territory; Walter E. Brewer, to cover northeastern Penn-

Wanted

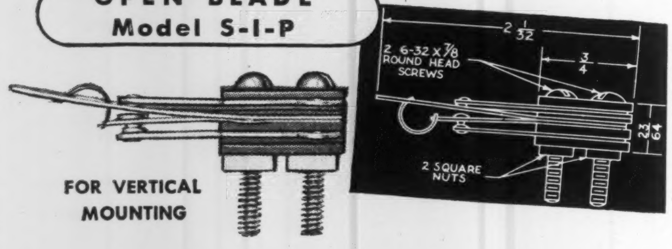
more upset forgings customers

If you're looking for better service in better forgings, Tube Turns is looking for you. With a complete and completely modern assembly of upsetters, including the world's largest—plus proved ability at developing tailor-made procedures for specific jobs—this experienced but young and progressive organization is ready to serve several more customers of the kind who want an extra something when it comes to the mass-production of forgings. Write or call Forgings Development Desk, Tube Turns (Inc.), Louisville 1, Ky. Wabash 7551.

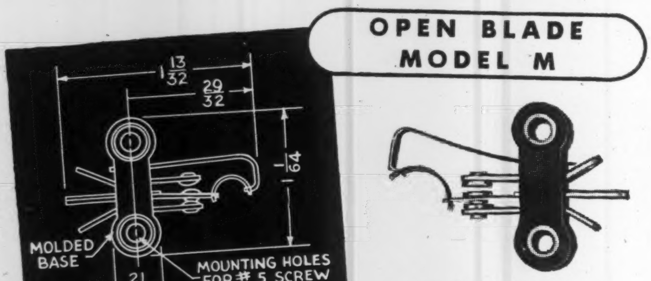
TUBE TURNS tt *Forgings for Industry*

DO YOU MOUNT *Vertical or* HORIZONTAL?

OPEN BLADE Model S-1-P



OPEN BLADE MODEL M



FOR HORIZONTAL MOUNTING

ACRO *Open Blade* SWITCHES

Simplify hundreds of problems

Whether you mount your switches vertically or horizontally you too should be able to improve product performance by adopting ACRO Snap-Action Switches. The flexibility of design and adaptation and the amazing performance records are keyed to ACRO's patented beryllium Rolling Spring construction. No other switch is made like it. No other switch performs like it. Yet the cost is low.

With better contact pressures, sturdy construction and positive fast action, ACRO Switches are virtually trouble-free. Many are built for millions of operations. Made in 1, 2 and 3 poles—single throw and double throw. Return types and set types with a wide range of operating pressures. Rated: 15 Amps. at 125 volts A.C. Illustrated above are but 2 of over 1500 models of open blade and inclosed type switches. Send your engineering details for a prompt answer.

ACRO ELECTRIC COMPANY

1311 Superior Avenue Cleveland 14, Ohio

sylvania; Frank J. Chadwick, the Borough of Manhattan; Lawrence E. Rossiter, eastern Connecticut, Rhode Island and southeastern Massachusetts; Neil E. Elphick, northern New Jersey and New York State as far north as Saugerties; Thomas D. Wilkinson, central New Jersey and Staten Island; and Alfred M. Hartley, southeastern Pennsylvania, southern New Jersey, Delaware, Maryland and Virginia.

◆

Recently released from the Signal Corps, Daniel J. Killfoile has joined the sales staff of Alloy Steel Products Co., Linden, N. J. Prior to entering the service he was a purchasing engineer for Chemical Construction Co.

◆

Howard B. Jones Co., 2460 West George St., Chicago 18, has been purchased by the Cinch Mfg. Corp. To be known as the Howard B. Jones Division, the plant will continue the manufacture of electrical connecting devices. Key men and women of the company have been retained.

◆

Opening of the Empire State Technical Section of its development and research division has been announced by The International Nickel Co. Inc. Gilbert L. Cox, metallurgical and chemical engineer, will be in charge of the new section which is located in the Genesee Valley Trust Bldg., Exchange and Broad Sts., Rochester 4, N. Y. The section will furnish technical information and assistance to industry in the state of New York excluding New York City, the Albany area and the Hudson River valley.

◆

To succeed E. B. Maire who resigned recently, Penn Electric Switch Co. of Goshen, Ind., has appointed Walter W. Lige as manager of the Chicago branch office at 520 North Michigan Ave. Formerly Mr. Lige was regional merchandise manager of Montgomery, Ward & Co.

◆

Connected with the company since 1920, L. E. Lynde has been named manager of the New England district of Westinghouse Electric Corp., with headquarters in Boston. He has succeeded F. L. Nason who will become a special representative of the company in the New England district. Also announced are the appointments of Sidney C. Palmer as Transportation and Marine Division manager, and F. S. Bacon as Central Station manager for the New England district.

◆

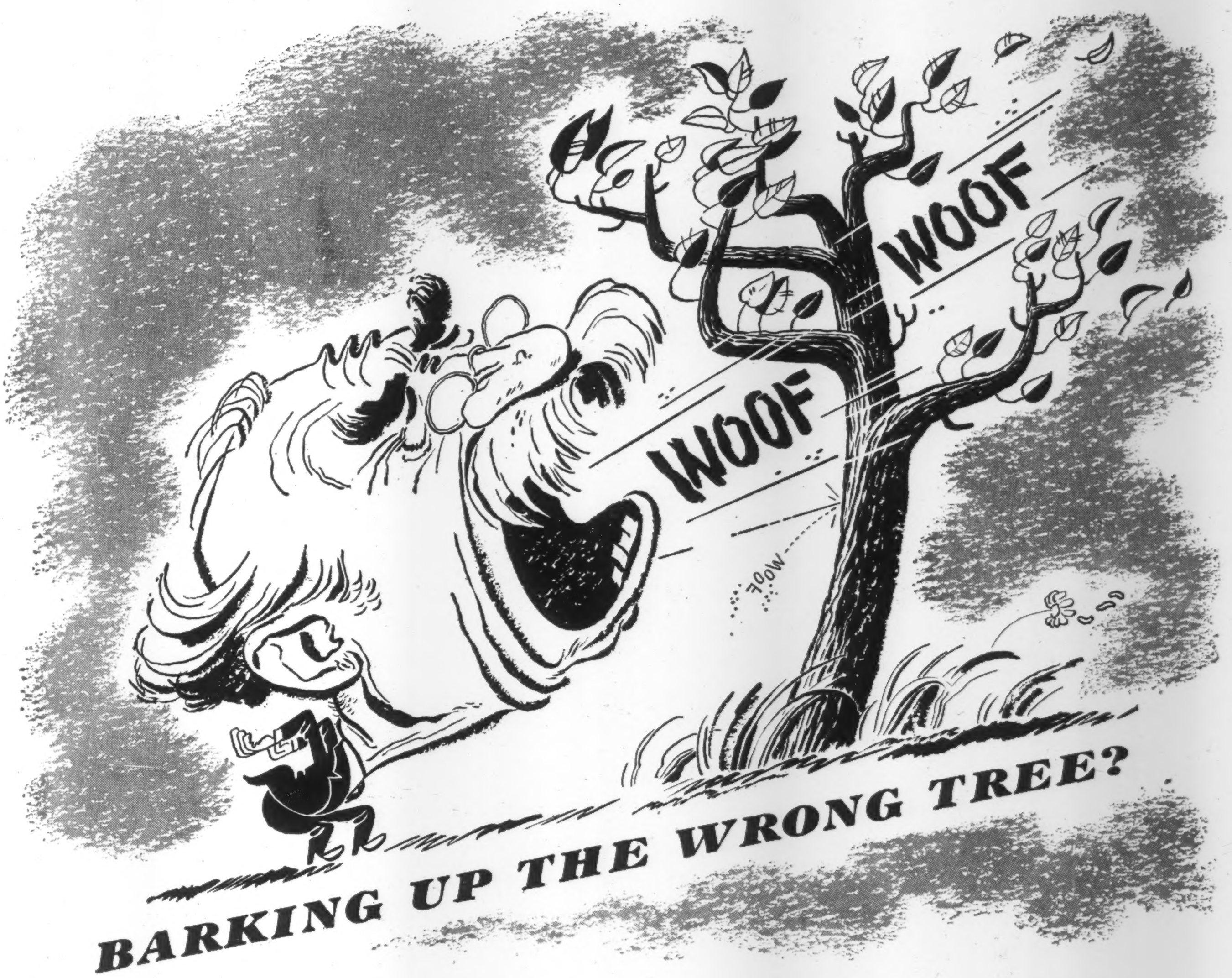
Ritchie Engineering Co., 4516 Wooddale Ave., Minneapolis 10, has been made exclusive sales representative in Minnesota and northwestern Wisconsin for Bound Brook Oil-Less Bearing Co., Bound Brook, N. J. All classes of the company's lubricant-retaining bearings will be handled by the new sales representative.

◆

Change of name has been announced by The American Gauge Co., Dayton 1, O. Henceforth it will be known as The American Gage & Mfg. Co. No change in personnel or location has resulted.

◆

Purchase of the Progressive Foundry Works Inc. of Rochester, N. Y., has been planned by American Brake Shoe Co. The new plant, to be operated as a division of the company, will be under the direction of W. T. Kelly Jr. who is also president of the Kellogg Division. Howard S. Van Billiard, president of


WOOF
WOOF
BARKING UP THE WRONG TREE?

TRY THIS SERVICE

Are you having a devil of a time designing a new process or machine? If so, it's possible some of your difficulties can be ironed out by the partial or complete use of Fluid Power.

There's where our service comes in.

We build the Hele-Shaw Fluid Power Pump. But that's only the beginning. Our engineers are geared to tackle the job of designing and developing complete Fluid Power Systems.

And we will follow through, if asked, on the assembly of all component elements—one responsibility all the way.

We urge you to use this service, backed by 30 years of Fluid Power experience, *now*. Call in our representative nearest you (see your classified directory under Hele-Shaw) or write us for details. No obligation.

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***Fluid Power* PUMPS**

FLUID-POWER *Systems*

OTHER Æ PRODUCTS:

Æ-TAYLOR AND PERFECT SPREAD STOKERS,
MARINE DECK AUXILIARIES, LO-HED HOISTS.

Æ

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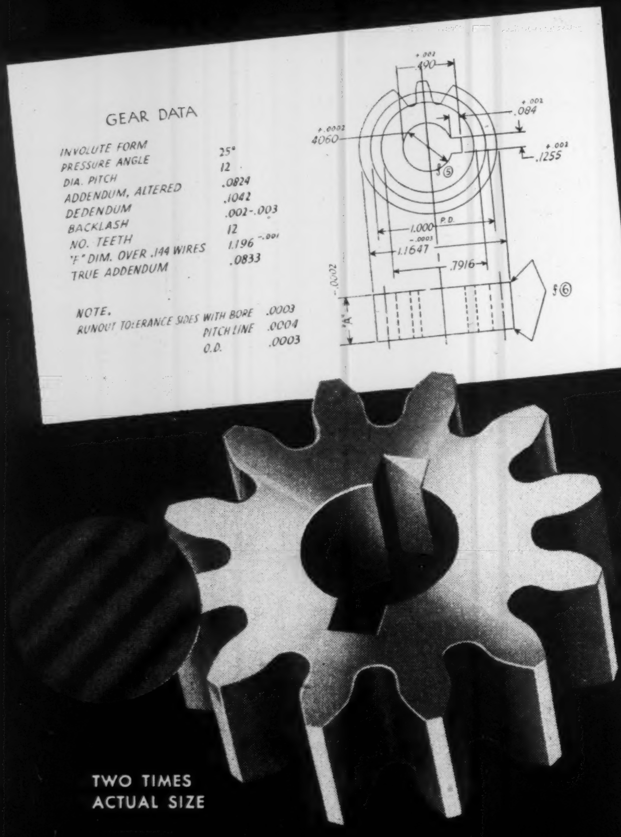
NO EXTRA COST FOR
Precision Gears
MADE THE McINTYRE WAY



GEAR DATA
INVOLUTE FORM 25°
PRESSURE ANGLE 12
DIA. PITCH .0824
ADDENDUM, ALTERED .1042
DEDENDUM .002-.003
BACKLASH 12
NO. TEETH 1.196 -.001
"F" DIM. OVER .144 WIRES .0833
TRUE ADDENDUM

NOTE.
RUNOUT TOLERANCE SIDES WITH BORE .0003
PITCH LINE .0004
O.D. .0003

TWO TIMES
ACTUAL SIZE

This is one size of the twin spur gears used in McIntyre Series 100 Metering Pumps which deliver ¼ to 3 gpm at pressures up to 1000 psi for oil or up to 200 psi for water at speeds up to 3600 rpm.

McINTYRE PRECISION GIVES YOU GEARS WITH TOLERANCES LIKE THESE: Look at the tolerances of the illustrated gear and consider how similar tolerances can improve the performance of your geared mechanisms: sides lapped within .00005″ of parallelism; bore .0002″; O.D. — .0003″; run out — sides with bore .0003″, pitch line .0004″, O.D. .0003″.

Tolerances like these are standard practice with the McIntyre Co., because our mass production machinery methods are *capable of holding vital dimensions to plus or minus .000025″ . . . surfaces flat to one light band.*

Whatever your requirements in spur gears up to 10″ O.D. in any material, you get them in tenth and split tenth tolerances by specifying McIntyre. Want estimates? The McIntyre Co., 300 Riverdale Ave., Newton 58, Mass.

THE McINTYRE CO.
(FORMERLY ZENITH ASSOCIATES)
PUMPS AND FLUID MOTORS

THE ULTIMATE IN PRECISION IDENTIFIED BY THE LIGHT BAND

Progressive Foundry, will remain in charge as manager. No changes in personnel are contemplated.

◆

During the next few months facilities will be installed by Cooper-Bessemer Corp. which will double its Grove City, Pa., foundry production.

◆

General Electric Co. has appointed D. F. Roloff, H. F. Pritchard and C. Stonehill to major sales positions in the newly created Specialty Transformer Division. Mr. Roloff, previously assistant sales manager of the old Specialty Transformer Section, has been named assistant manager of the new division. Mr. Pritchard and Mr. Stonehill have been made sales managers of the General Purpose Components Section and the Lighting Components Section, respectively.

◆

Previously sales manager, Jack L. Carmitcheal has been elected vice president in charge of sales for the Lincoln Engineering Co., St. Louis, manufacturer of lubrication equipment for automotive, agricultural implements and industrial applications.

Meetings and Expositions

June 12-15—
American Society of Mechanical Engineers. Oil and Gas Power Division meeting to be held at Milwaukee. Additional information may be obtained from headquarters of the society at 29 West 39th St., New York. C. E. Davies is secretary.

June 17-20—
American Electroplaters Society. Annual convention to be held at Hotel William Penn, Pittsburgh. Additional information may be obtained from headquarters of the society at 93 Oak Grove Ave., Springfield, Mass.

June 17-20—
American Society of Mechanical Engineers. Semiannual meeting to be held at Hotel Statler, Detroit. C. E. Davies, 29 West 39th St., New York is secretary.

June 21-22—
American Society of Mechanical Engineers. Applied Mechanics Division meeting to be held in Buffalo. Additional information may be obtained from headquarters of the society at 29 West 39th St., New York. C. E. Davies is secretary.

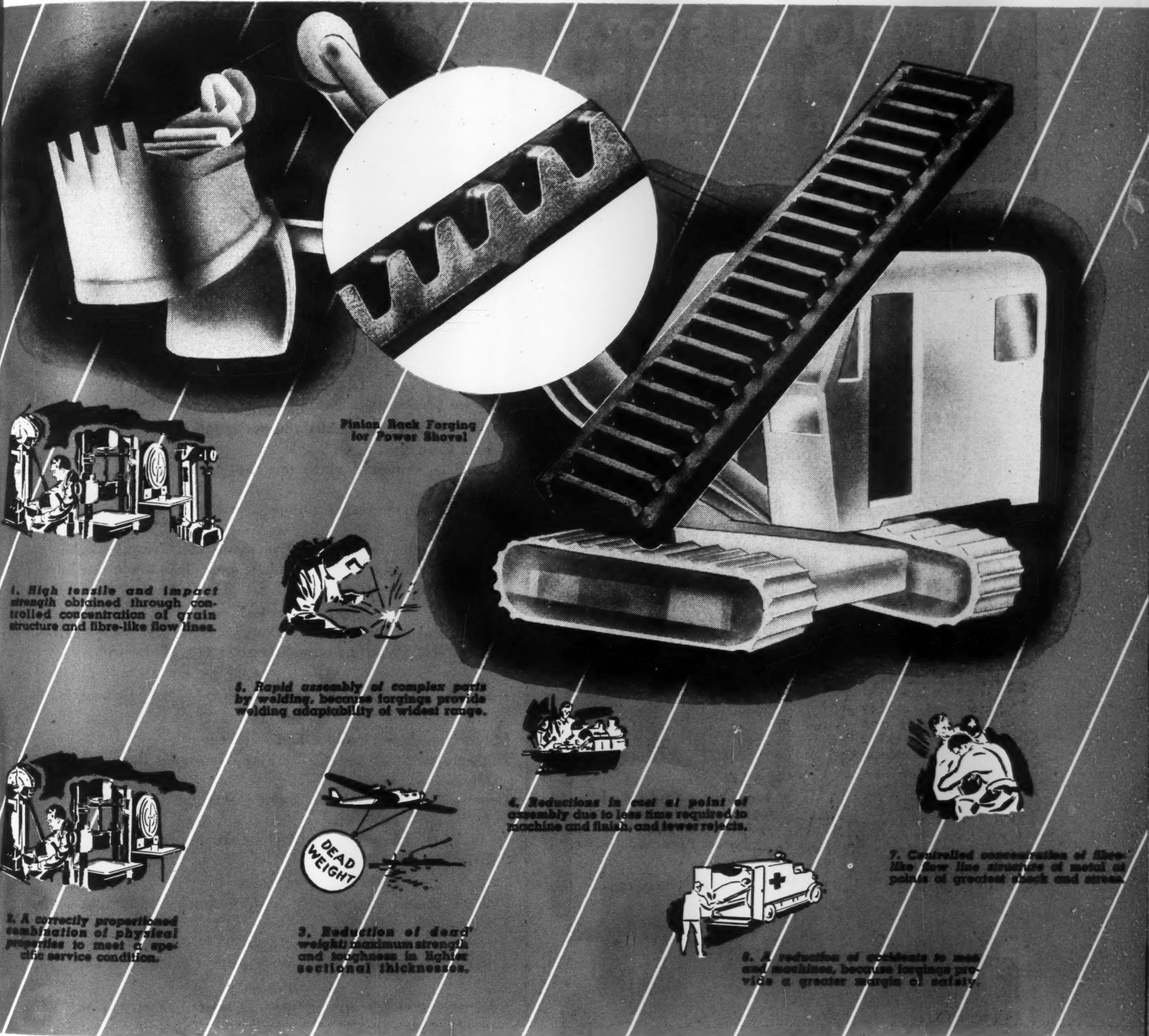
June 24-28—
American Institute of Electrical Engineers. Summer convention to be held at Hotel Statler, Detroit. H. H. Henline, 33 West 39th St., New York 18, is secretary.

June 24-28—
American Society for Testing Materials. Forty-ninth annual meeting and seventh exhibit of testing apparatus and related equipment to be held at Buffalo, N. Y. Additional information may be obtained from headquarters of the society at 260 South Broad St., Philadelphia 2. C. L. Warwick is secretary.

June 25-26—
National Warm Air Heating and Air Conditioning Association. Midyear annual convention to be held at Edgewater Beach Hotel, Chicago. George Boeddener, 145 Public Square, Cleveland 14, is managing director.

Aug. 22-24—
Society of Automotive Engineers Inc. National West Coast transportation and maintenance meeting to be held at New Washington Hotel, Seattle. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

MAXIMUM METAL QUALITY ISN'T ALL THAT YOU GET IN . . Forgings



Pinion Rack Forging for Power Shovel

1. *High tensile and impact strength* obtained through controlled concentration of grain structure and fibre-like flow lines.

2. *A correctly proportioned combination of physical properties* to meet a specific service condition.

3. *Reduction of dead weight;* maximum strength and toughness in lighter sectional thicknesses.

4. *Reductions in cost at point of assembly* due to less time required to machine and finish, and fewer rejects.

5. *Rapid assembly of complex parts* by welding, because forgings provide welding adaptability of widest range.

6. *A reduction of accidents to men and machines,* because forgings provide a greater margin of safety.

7. *Controlled concentration of fibre-like flow line structure of metal at points of greatest shock and stress.*

Buyers of your equipment expect two things of it: that it will perform up to their expectations; that it will operate efficiently under whatever unusual service condition exists in the case of a particular user. *In forgings, metal quality can be developed to the exact degree required to meet a specific service condition.* Design parts to obtain the utmost tensile and impact strength, toughness and fatigue resistance from the fibre-like flow line structure inherent in wrought metals, of which forgings outperform other products. Forgings offer many advantages beyond those called for in the specification.

A recheck of every stressed part, as well as simple handles and levers, often reveals opportunities to improve the performance of a product, to reduce cost of machining and finishing, to speed up assembly. Consult a forging engineer while the part is in the design stage. He will show you how forging develops maximum metal quality.

METAL QUALITY
HOT WORKING IMPROVES PROPERTIES OF METAL
TOPICS

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☐ Booklet on "Metal Quality—Hot Working Improves Properties of Metal."
☐ "Drop Forging Topics," issued at 60 day intervals.

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